Calhoun: The NPS Institutional Archive
DSpace Repository

Theses and Dissertations 1. Thesis and Dissertation Collection, all items

1958

Lubrication study of an oscillating bearing.

Nelson, George G.

Monterey, California: U.S. Naval Postgraduate School

Downloaded from NPS Archive: Calhoun

Calhoun is the Naval Postgraduate School's public access digital repository for research materials and institutional publications created by the NPS community. Calhoun is named for Professor of Mathematics Guy K. Calhoun, NPS's first appointed -- and published -- scholarly author.

Dudley Knox Library / Naval Postgraduate School
411 Dyer Road / 1 University Circle
Monterey, California USA 93943

http://www.nps.edu/library

NPS ARCHIVE
1958
NELSON, G.

LUBRICATION STUDY OF AN OSCILLATING BEARING

George G. Nelson

DUDLEY KNOX LIBRARY
NAVAL POSTGRADUATE SCHOOL
MONTEREY CA 93943-5101

LUBRICATION STUDY OF AN OSCILLATING BEARING

* * * * * * * * *

George G. Nelson

LUBRICATION STUDY OF AN OSCILLATING BEARING

by

George G. Nelson

Lieutenant, United States Navy

Submitted in partial fulfillment of
the requirements for the degree of

MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

United States Naval Postgraduate School
Monterey, California

1958

NPS ARCHIVE
1958
NELSON, G.

~~Thesis~~
~~N362~~

LUBRICATION STUDY OF
AN OSCILLATING BEARING

by

George G. Nelson

This work is accepted as fulfilling
the thesis requirements for the degree of

MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

from the

United States Naval Postgraduate School

ABSTRACT

This investigation covers the regimes of operation of an unidirectionally loaded sleeve bearing. The particular journal studied was a hardened steel member oscillating through 30°, at various loads and speeds in a hardened steel bushing. From the results it was concluded that there were possibly four regimes of operation. The coefficient of friction with constant load and increasing speed was found to decrease and then to increase. Three methods were used in the investigation to determine the lubricating characteristics: friction measurement, oil film resistance, and radioactive tracer.

The experimental work from which this paper is written was conducted at the United States Naval Postgraduate School in the Fall and Winter of 1957.

PREFACE

The oscillating bearing typified by the piston pin bearing in an internal combustion engine represents a good example of non-steady state lubrication problems. No analytical solution to this problem was available, nor was the simplified problem of a constant unidirectionally loaded oscillating bearing available. The past experimental work has been limited, and engineers have been required to make assumptions on strictly negligible evidence of what occurs during operation. The problem has not been dealt with more extensively, possibly because few cases of failure have presented themselves in the form of unsuccessful lubrication in practice in the case of the piston pin which is the most common application. The varying load in this particular case gives a pumping action to the bearing, and consequently a building up of the oil film.

Without the encouragement and ingenuity of Professor Ernest K. Gatcombe of the Mechanical Engineering Department, and the advice of Professor Gilbert F. Kinney of Metallurgy and Chemistry Department, in the plating techniques involved, little would have been accomplished. The machine shop of the Naval Postgraduate School, and Mr. Ruel P. Kennicott must be particularly praised for the excellent job of fabrication done on the driving mechanism and miscellaneous adaptors for converting the Milling Machine to a bearing test machine for this investigation.

TABLE OF CONTENTS

Section	Title	Page
1.	Introduction	1
2.	Statement of Problem	2
3.	Description of Apparatus	3
4.	Radioactive Plating	10
5.	Instrumentation	13
6.	Calibration	16
7.	Results	19
8.	Description of Graphical Results	23
9.	Discussion of Results	36
10.	Bibliography	41
Appendix I	Lubricating Oil Specifications	42
Appendix II	Tabulated Data	43
Appendix III	Calibration Curve for Resistance Measurements	60

LIST OF ILLUSTRATIONS

Figure		Page
1.	Testing Machine	4
2.	Schematic Sketch of Driving Mechanism	5
3.	Front View of Milling Machine	6
4.	Bearing Assembly, Journal, Loading Assembly, and Strain Gage Element	8
5.	Plating Set Up	12
6.	Strain Gage Element and Pulley for Calibration	14
7.	Wiring Diagram of Resistance Measuring Section and Shaft Position Transmitter	17
8.	Block Electrical Wiring Diagrams	18
9.	Graphical Results at Constant Load	25
10.	Graphical Results at Constant Load	26
11.	Graphical Results at Constant Load	27
12.	Graphical Results at Constant Load	28
13.	Graphical Results at Constant Load	29
14.	Graphical Results at Constant Load	30
15.	Graphical Results at Constant Rate of Oscillation	31
16.	Friction Traces Versus Time at Constant Speed of 120 opm	32
17.	Friction Traces Versus Time at Constant Speed of 120 opm	33
18.	Oscilloscope Traces of Oil Film Resistance at Constant Speed of 120 opm	34
19.	Typical Oscilloscope Traces	35

TABLE OF SYMBOLS AND ABBREVIATIONS

General

^{o}F	Temperature degrees Fahrenheit
Kg	Kilograms
N	Oscillations per minute
P	Load in pounds per square inch
p.s.i.	Pounds per square inch
o.p.m.	Oscillations per minute (a complete cycle)
r.p.m.	Revolutions per minute
SSU	Saybolt Second Universal
Z	Viscosity in Reyns.

Chemical

ml	Milliliter
HCl	Hydrochloric Acid
$FeCl_3.6H_2O$	Hydrous Ferric Chloride
KCl	Potassium Chloride
N	Normality

INTRODUCTION

The oscillating bearing has been a much neglected subject of research, however, two notable papers have been written on this type bearing. Fogg and Jakeman[3] reported on the variation of friction at the middle of the stroke and noted a decrease of the coefficient of friction with increasing rate of oscillation. Their experiments were primarily concerned with speeds up to 15 opm, and they used a special "grooved bush" bearing. Barwell and his associates[1] in 1955 reported upon an extensive series of experiments which was labeled by Hersey[4] a report likely to take high rank among studies of its kind.

Among Barwell's conclusions, was, that it was possible to obtain under proper conditions lubrication analogeous to hydrodynamic in bearings undergoing oscillatory motion. He also noted that the variation of friction during the stroke fell into three general categories: (a) sinusoidal (b) constant value (c) a special form in which the friction was larger at both ends of the stroke. These forms appear with light load and high speed for the sinusoidal to heavy loads and slow speeds for the special form. Barwell's tests were run on leaded bronze, lead base white metals*, and tin base white metal bearing materials with a mild steel journal.

---

* Used for the friction tests mentioned above.

STATEMENT OF PROBLEM

The intention of this investigation was to determine the character of the lubrication in a unidirectionally loaded full journal bearing undergoing oscillation. To eliminate the effects of wear and plastic flow, it was decided to use hardened steel journals and bushings. No attempt was made to improve the lubrication characteristics of the bearing. Only a central oil inlet hole was used. The methods of measurement used to obtain information were: friction measurement, oil film electrical resistance, and radioactive tracer technique using Fe^{59} plated on the journal.

DESCRIPTION OF APPARATUS

In order to obtain a good foundation for our test apparatus, a surveyed Universal Milling Machine of the Column and Knee type was used. See Fig. 1. This not only provided a solid foundation with mass enough to absorb most of the minor vibrations, but avoided problems of alignment by mounting the entire driving mechanism on the table. By use of two Arbor Yokes on the arms to support either end of the test journal, good support was given to the journal. Rotary motion was obtained from a Vickers 3/4 horsepower Hydraulic Transmission, driven by a 1/2 horsepower 110 volt electric motor. This particular drive has a speed range from 0 to 1725 r.p.m. Oscillation through a total angle of 30 degrees was obtained with the use of an eccentric, crank arm and crank positioned so that symmetrical motion approaching sinusoidal oscillation was achieved. This mechanism proved satisfactory for speeds up to 700 r.p.m. by using counterweights for static balance of the eccentric and part of the crank arm. Because of the inherent unbalance of this apparatus (critical speed of 900 r.p.m.), test runs were limited to speeds up to 600 r.p.m., except for qualitative information at higher speeds. Schematic diagram of the mechanism is shown in Fig. 2. Photograph of the mechanism is shown in Fig. 3.

The test journals were commercially obtained hardened centerless ground one inch diameter piston pins. An insert was forced into one end of the pin and machined concentrically for attachment

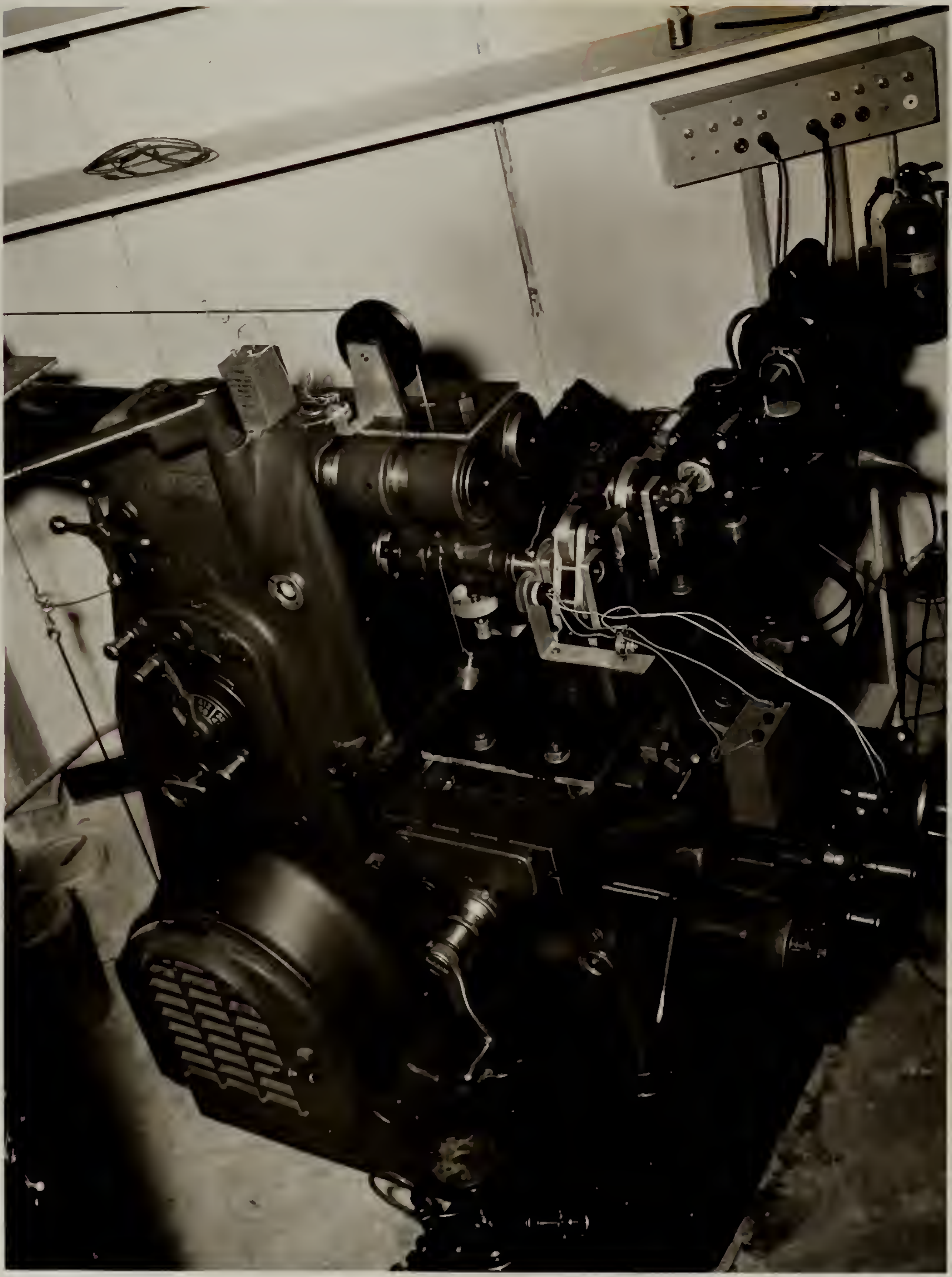
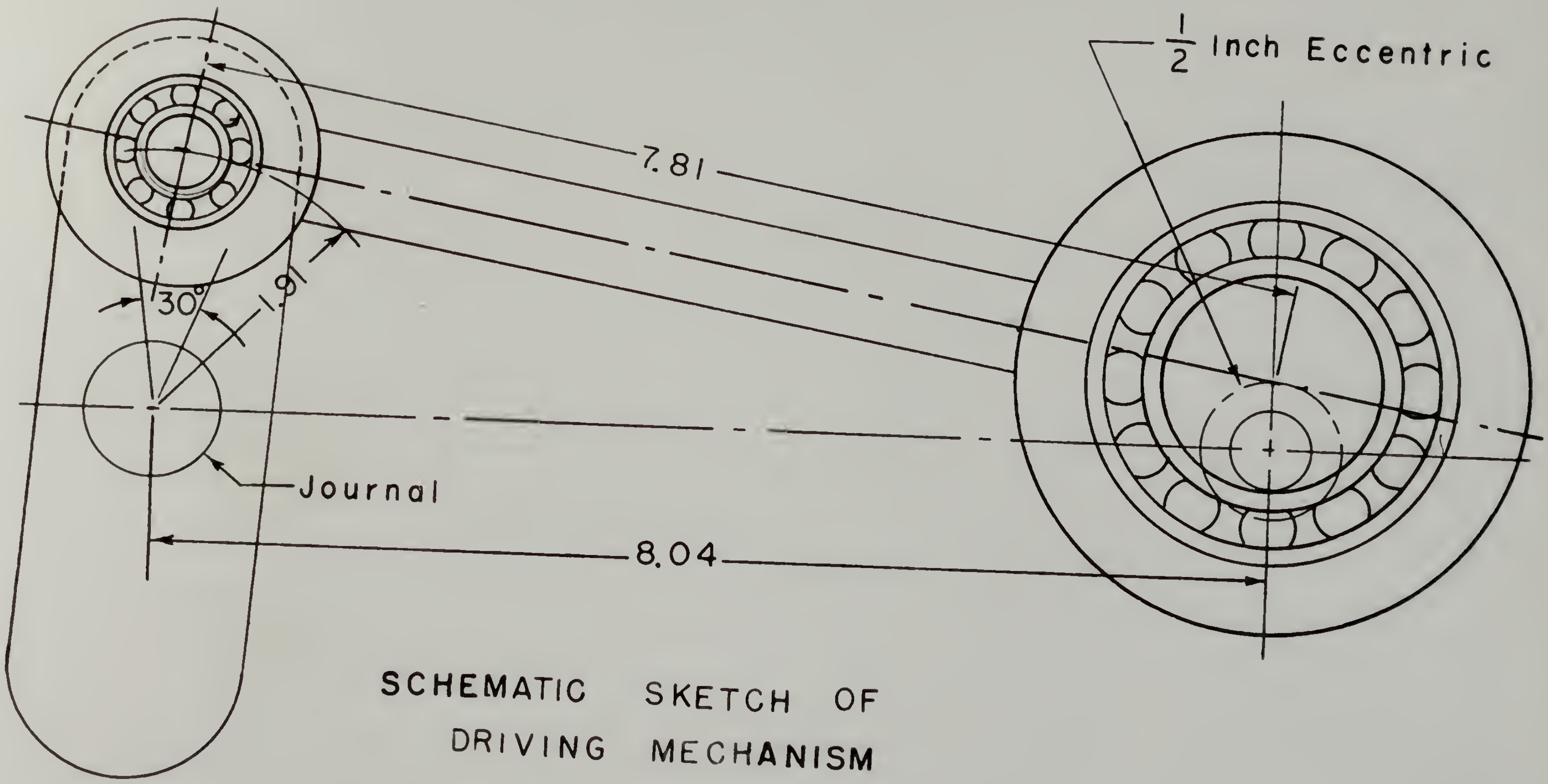


Figure 1 - Testing Machine

Apparatus mounted on Milling Machine showing calibration of strain gage element in progress.



SCHEMATIC SKETCH OF
DRIVING MECHANISM

FIGURE 2

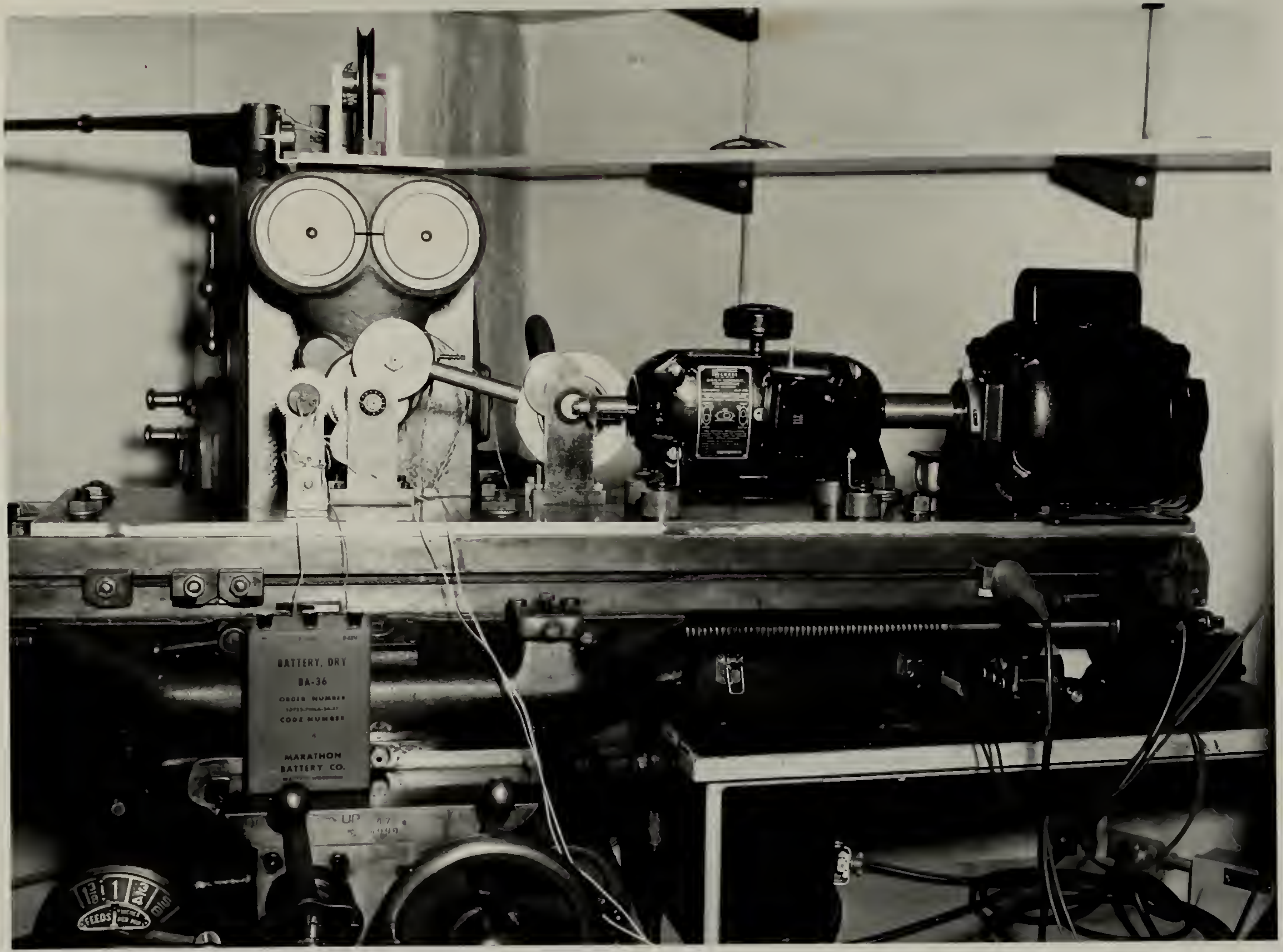


Figure 3 - Front View of Milling Machine

Drive mechanism, Vickers Hydraulic Transmission, electric motor and shaft position transmitter geared to crank arm are shown.

to the crank arm. The journal ran in two needle bearings mounted in the arbor yoke of the milling machine.

The bushing was one inch long of ground and hardened steel, and had an interference fit with a brass bearing holder assembly. Two O-ring gaskets were used between the bearing holder and the bushing to prevent oil leakage. Load was applied to the bushing by a 1/8-inch 2,000 pound test flexible steel cable attached to the bearing assembly. The bearing holder was split to enable interchange of bushings. The 1/8-inch cable was led over two bakelite pulleys to a loading pan. The entire assembly was thus kept electrically insulated from the Milling Machine. See Fig. 4 for layout of Bearing Assembly.

Oil was fed to the bearing through flexible Poly Vinyl Chloride, "Tygon", tubing from a Gage Tester which provided a constant pressure of three p.s.i. at the bushing. The bushing had a single 1/8-inch oil inlet hole in the middle of the loaded area.

The lubricating oil used throughout these experiments was Military Symbol 3050, a general purpose lubricant with no additives except a pour point depressant. The viscosity at 130°F is 45-55 SSU. Complete specifications will be found in Appendix I.

The strain gage element served two purposes: first, to prevent rotation of the bushing, and secondly, to measure the torque transmitted from the journal to bearing assembly. It consisted of a 5/8-inch steel tube .021 inches thick and .3 inches long, plus mounting accessories mounted axially parallel to the journal. Two S-R type A-18 electrical resistance strain gages were cemented

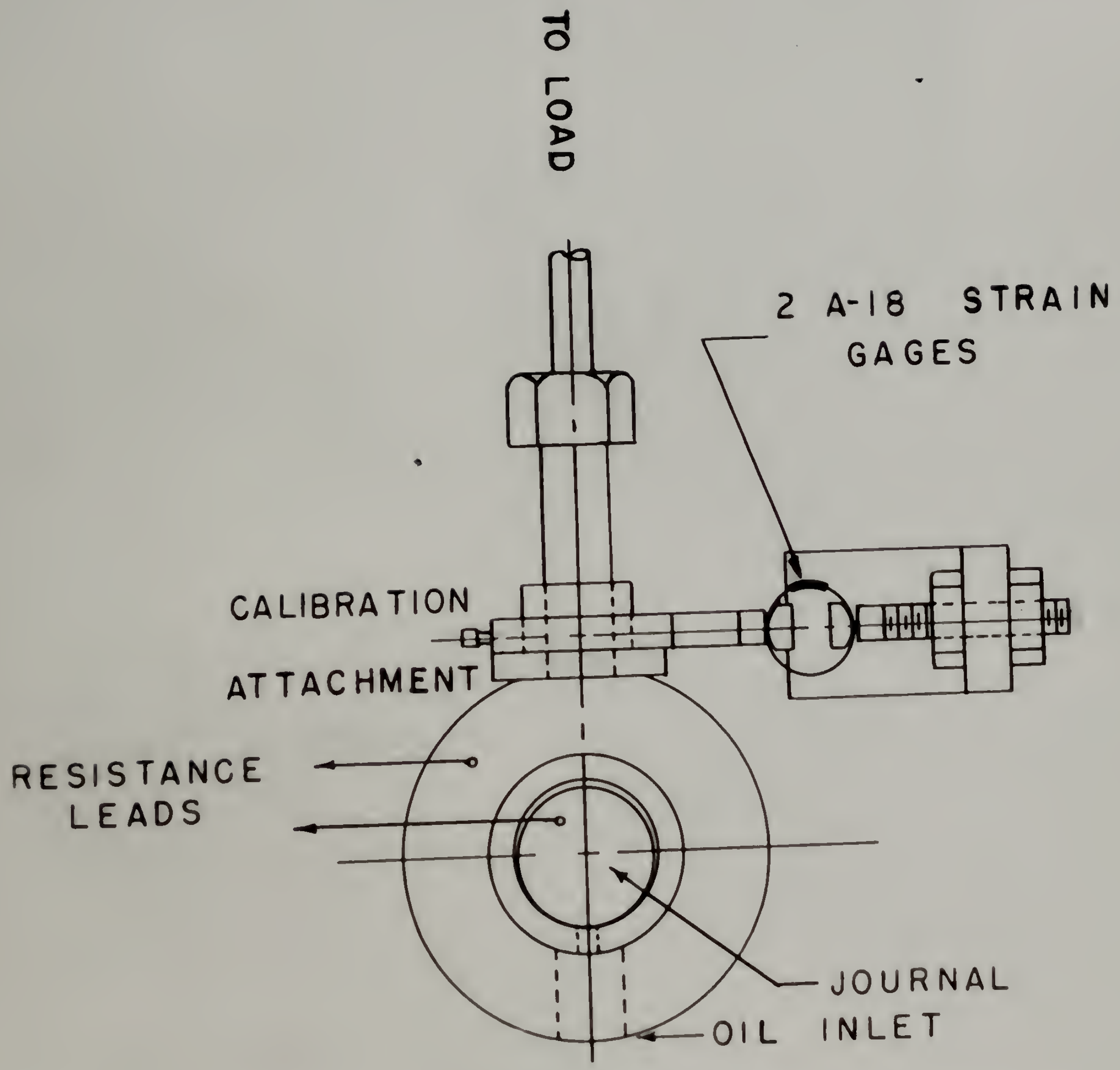


FIGURE 4 BEARING ASSEMBLY, JOURNAL, LOADING ASSEMBLY, AND STRAIN GAGE ELEMENT

opposite each other on the tube. This gave a two-fold magnification of strain readings. A lucite bushing was machined for a snug fit over the bearing assembly for insulation purposes, and one end of the strain gage element was fitted snugly over this bushing. The other end of the strain gage element was attached by two nuts, for adjustment, to a special attachment screwed into one of the stud holes on the arbor yoke. The tube was attached to each end of the element by use of a split 3/16-inch diameter bar bolted on each end. See Fig. 4

The shaft position transmitter consisted of a linear potentiometer of 1,000 ohms which was geared to the crank arm as shown in Fig. 4. A 22½ volt dry cell battery was connected across this resistance and the variation of voltage at the center tap of the potentiometer gave a signal which was directly proportional to shaft position.

RADIOACTIVE PLATING

The radioactive isotope Fe^{59} was obtained as Ferric Chloride ($Fe\,Cl_3$) in a .495 N solution of HCl. Three millicuries were received in 2.9 ml at a concentration of .35 milligrams of iron per milliliter. It was decided that plating should be accomplished at room temperature, therefore a Ferrous-Potassium Chloride bath suggested by Almo D. Squitero in "Metal Finishing Guidebook-Directory"[7] was used as a guide. The primary problem in plating was the reduction of the Ferric ion to the Ferrous condition. Several methods were attempted, but were found unsatisfactory. Finally it was found that the best method for reduction of small quantities of solution with a large surface to volume-ratio could be found by adapting methods from Analytical Chemistry. The method adapted to this particular problem was the Jones Reductor[6].

Normal procedure was used in preparing the reductor. A burette was used for the column. Reduction was carried through with the ferric ion in a solution of HCl rather than Sulfuric acid solution. The base plating solution was made up of 404 grams per liter of $Fe\,Cl_3 \cdot 6H_2O$ and 180 grams per liter of $K\,Cl$ in .5 N HCl solution. 12.1 ml of this solution was combined with 2.9 ml of the radioactive solution and then reduced to the ferrous state for plating.

A combination anode-container was fabricated out of iron pipe. A rubber stopper was used for water tight closure at the bottom. Spacer rings were fabricated out of "lucite" and placed at the bottom and top of the container to serve the dual purpose

of concentrically locating the shaft for uniform plating, and to keep from needlessly plating 5/8-inch of the bottom end of the journal where it ran in the supporting needle bearing. An area approximately 1½ inches long was plated with the 15 ml of electrolyte. The plating procedure used was to apply a strike at 3 open circuit volts (resulting in 1.4 volts and 5 amperes) for 30 seconds, and then using a periodic reverse of 20 second plate and 5 second deplate with a current of .5 amperes. The resulting plate was .0001 to .0002 inches thick and gave a good reproduction of the surface from a visual inspection.

The journals were prepared for plating by a degreasing in a vapor bath of Dichloroethlene immediately prior to plating, and then placing in a 1 N solution of HCl for approximately 3 minutes before striking in the plating bath.

Complete plating procedure:

1. Degreasing bath. Thirty minutes.
2. Drying.
3. 1 N HCl rinse.
4. Water rinse.
5. Plate (immediately after reducing electrolyte).
6. Water rinse.
7. Alcohol rinse.
8. Drying.

A photograph of the plating apparatus, Jones Reductor, and last two rinses in shown in Fig. 5.

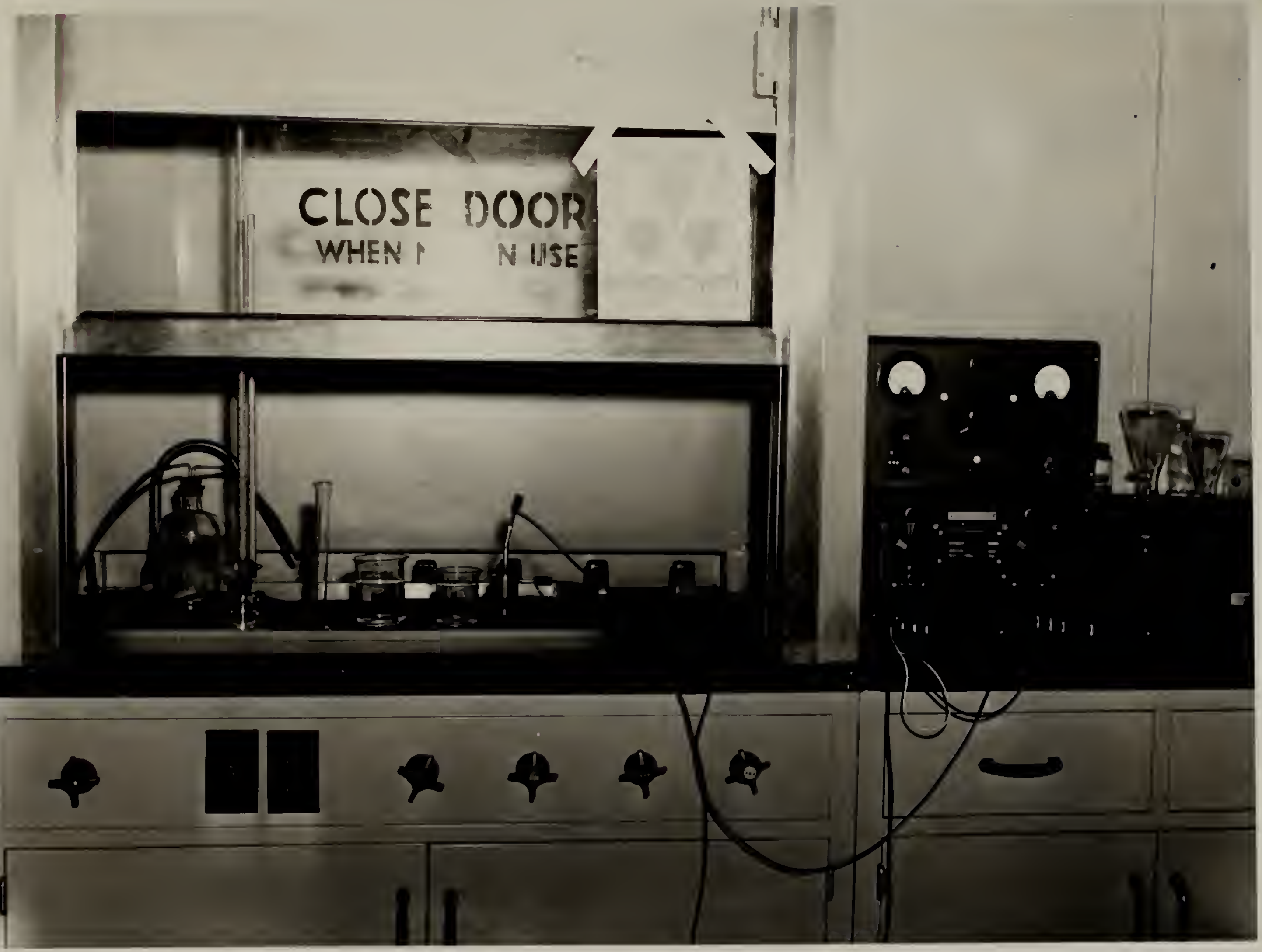


Figure 5 - Plating Set Up

Jones Reductor is shown at extreme left. Anode-Container is in center of picture with two beakers for water rinse and alcohol rinse. Current Source, Periodic Reverser and Volt and Ampmeters are at extreme right.

INSTRUMENTATION

The strain gages mounted on the tube in the strain gage element, shown in Figs. 4 and 6, provided the means of determining the frictional torque transmitted from the journal to the bearing. The maximum sensitivity of this arrangement when used in connection with a Brush Strain Analyzer and Recorder was such that under light loads each line of recorder paper spacing represented .0366 pound-force or .11 pounds of frictional force at the bearing surface.

The strain gage element unfortunately had a natural frequency of 56 cps determined by placing a dynamic load on the element when the equipment was not in operation and no load on the bearing. This natural frequency appeared in the recorder traces. It might also be hypothesied that these oscillations could have been the result of a slip stick phenomena in the bearing. Because there was no load on the bearing when the natural frequency was checked, this possibility seems highly unlikely. It is felt that the traces show only the result of the natural frequency and the dampening. Block wiring diagrams of this equipment are shown in Fig. 6.

A notch type filter was built and placed in the circuit to electrically dampen out the frequency, but discarded when it was found that distortion to the basic strain signal was introduced. When the friction trace was placed on the oscilloscope against shaft position, the natural frequency gave an even more distorted impression of the actual friction.

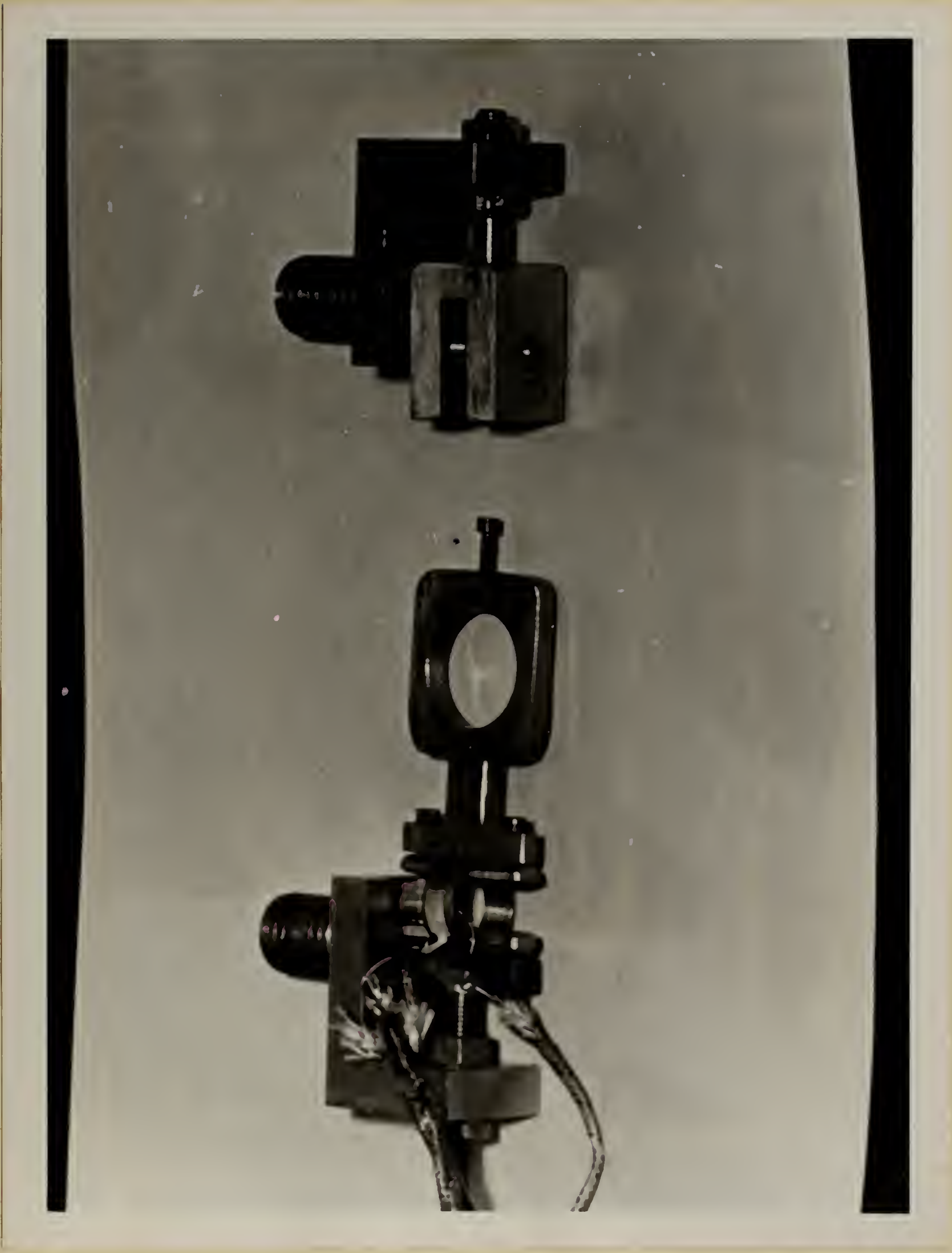


Figure 6 - Strain Gage Element and Pulley for Calibration

Courtney-Pratt and Tudor[3] found that the electrical breakdown of an oil film could be avoided if the potential drop across the film was less than .3 volt. Because of this, direct current resistance measurements were made between bushing and journal with .2 volt with infinite resistance. The voltage source was a standard 1½ volt dry cell battery. By using a resistance pot .2 volts was obtained. A visual representation of the resistance was obtained by placing leads from the bearing and journal across the terminals of an oscilloscope, and noting the displacement of the trace on the scope. This oscilloscope reading was actually a reading of the potential across the oil film. Note wiring diagram in Fig. 7.

CALIBRATION

The calibration of the strain gage was accomplished prior to every reading by placing a known weight on the end of a string lead over a pulley, and attached to the strain gage element, as shown in Fig. 7. This amounted to placing a static load on the Strain Gage Element. It was found that the load placed on the element was directly proportional to the value of strain recorded. These calibrations were made with the equipment in operation with the load on the bearing, thus superposing a static load on the dynamic load. See Fig.11 for typical results.

For determining the equivalent resistance of the displacement on the oscilloscope of the potential across the oil film, the leads were detached from the Journal and Bearing Holder, and attached to known resistance. The value of the resistance was plotted against the displacement on the Oscilloscope. This was accomplished immediately prior to making readings, and checked upon completion. The calibration curve is in Appendix III.

Since a dual beam oscilloscope was used, some qualitative information was derived by observing simultaneously, friction torque and oil film resistance against shaft position.

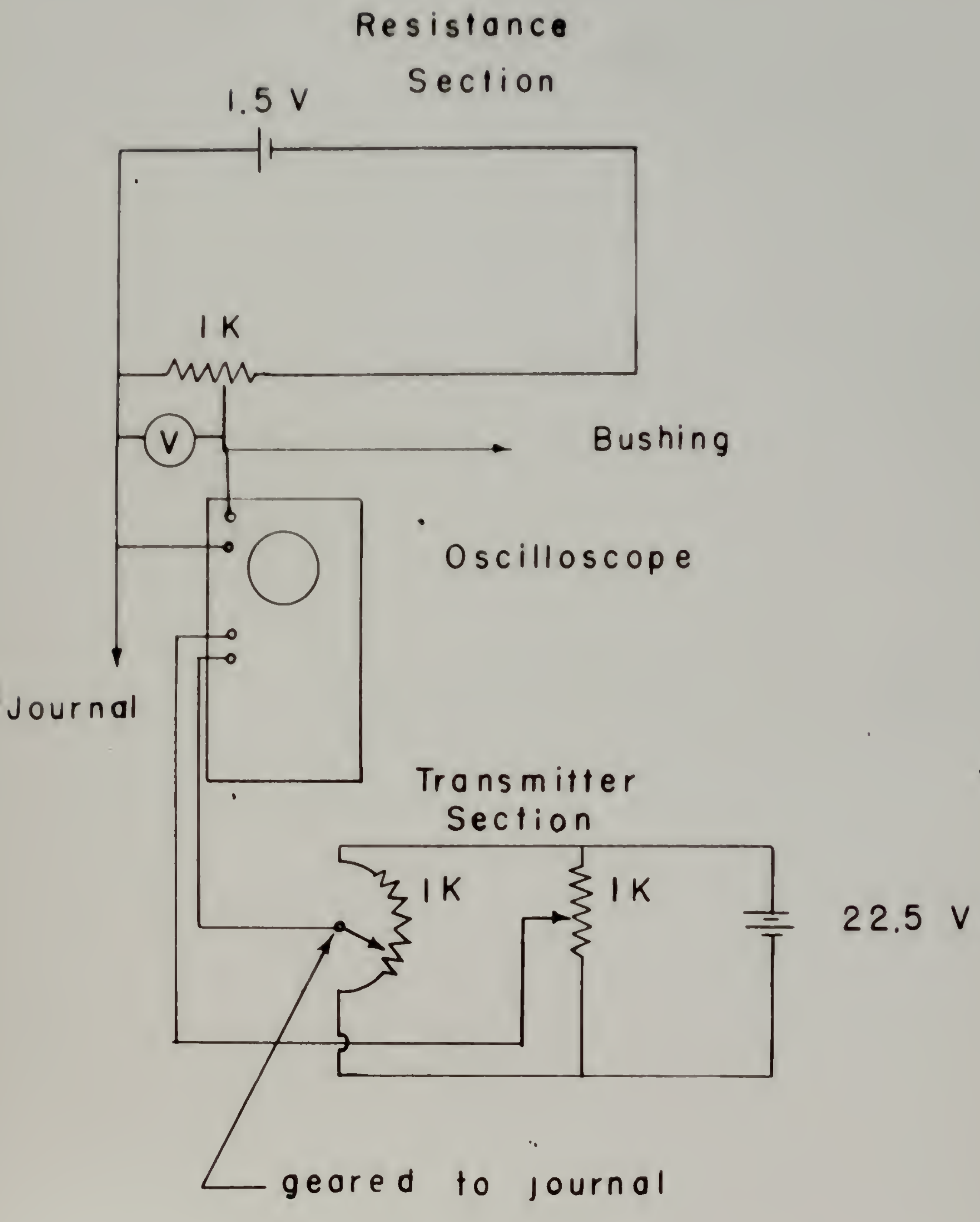
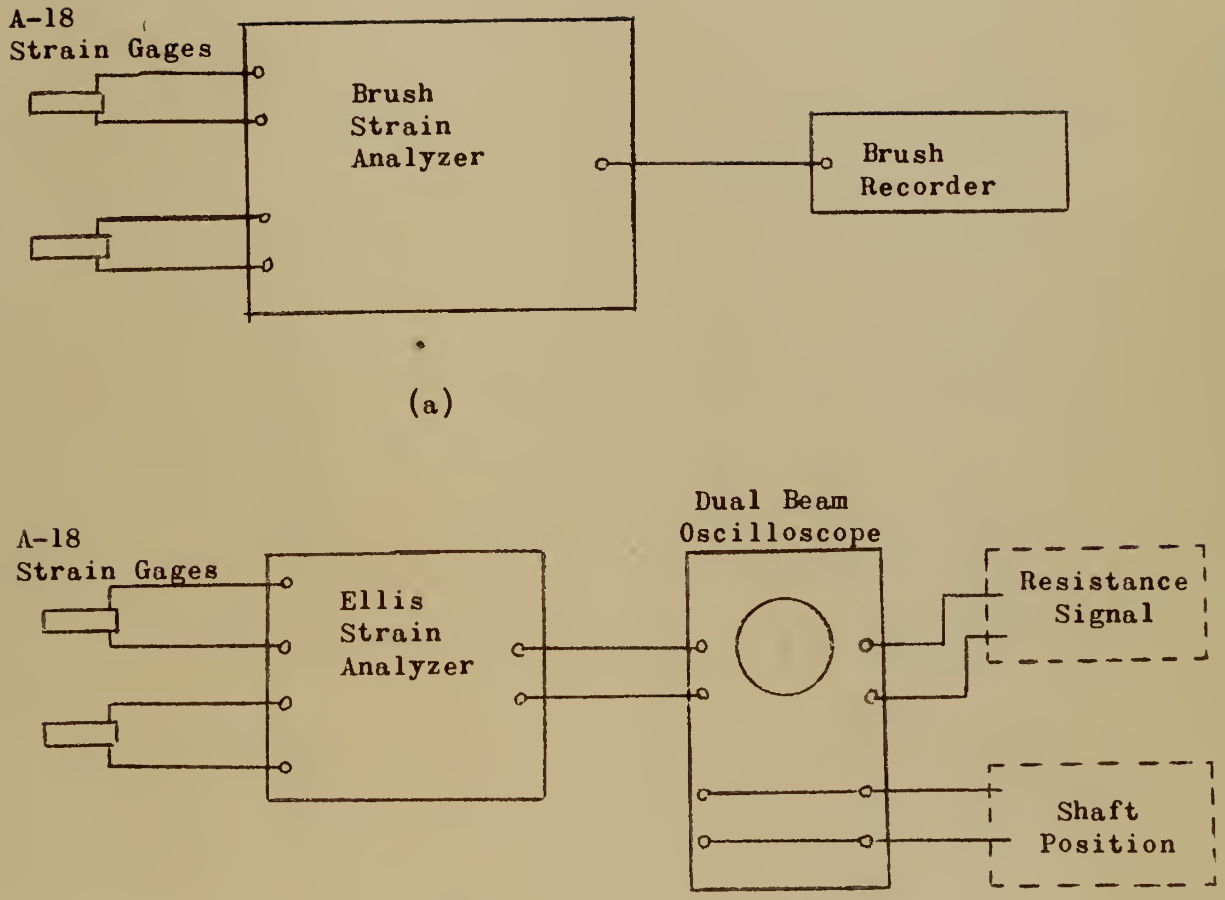


FIGURE 7. WIRING DIAGRAM OF RESISTANCE MEASURING AND SHAFT POSITION TRANSMITTER WIRING.



(a)

(b)

Figure 8 - Block Wiring Diagram

(a) Shows layout used for taking all quantitative data on friction.

(b) Shows layout used for taking qualitative data and for all data on resistance.

RESULTS

The investigation of the torque transmitted to the bushing by the journal, by means of partial or complete oil film viscous drag, hereafter referred to as friction, was conducted at speeds varying from 28 opm to 600 opm, and with loads varying from 24.6 to 195.6 p.s.i. (which is actually equivalent to total load, as the projected bearing area is one square inch). Results of these experiments are on two different journals and bushings. Results are plotted graphically in Figs. 9 through 14, with constant load lines plotted on coefficient of friction versus oscillations per minute coordinates. The coefficient of friction plotted was the value at the middle of the stroke irregardless of the form of the friction wave at any particular load or speed. Other values, such as the maximum value, minimum value, value at the commencement or completion of each half oscillation, or an average could have been used.

It was felt that the frictional value would have more physical significance at the middle of the stroke, as speed of the shaft was a maximum, and acceleration was approximately zero because of sinusoidal oscillation. Other values might have been less representative. Barwell used a maximum value of friction and plotted it against the ratio ZN/P . His work had direct applicability where this maximum value might be necessary. The use of dimensionless ratios, such as those of the Sommerfield work, would have been confusing, because there is some doubt as to their applicability in this case.

Determination of the rate of oscillation at rates over 90 opm was made using a calibrated tachometer; slower rates were measured directly by timing. Before any friction measurements were made, care was taken that equilibrium had been achieved, and calibration of the strain gage element was checked. Fig. 14 shows typical friction calibration traces from the recorder on the 145.5 p.s.i. load run.

Fig. 15 shows a set of runs made at 120 opm with various loads. Friction traces were recorded on the Brush Recorder simultaneously with the photographing of the oscilloscope trace of oil film resistance versus shaft position signal from the shaft position transmitter. A polaroid camera was used for this operation; each photograph actually represents four traverses across the scope. Calibration curve of resistance versus lines on the oscilloscope will be found in Appendix III. This calibration was made immediately prior to taking these runs, and checked immediately upon completion. Plotted values of electrical resistance are maximum values irregardless of position. No method was available to determine an equivalent thickness of the oil film separation that these resistance readings represented. Fig. 18 illustrates the pictures taken, and the friction traces made are shown in Figs. 16 and 17.

A Brush Surface Analyzer was used to determine the surface roughness value before and after operation of the bushing and journal. Results showed both journals had rms value of approximately 6 - 8 microinches prior to operation, and 7 - 9 microinches

after operation. Bushings showed rms readings of 10 - 12 microinches before, and 11 - 13 microinches after operation. There was no noticeable taper on either journal in the bearing area, but bushing number one used on the runs indicated in Figs. 9 to 11 had a taper of .0002 inches from end to end. No indications of wear were shown by measurement on either the journal or bushing in either case. The total clearance was .0015 inches on set number one, and .0009 inches on set number two. Rockwell "C" Hardness Readings were 62 on the journal and 53 on the bushing.

A check was made while running-in a journal and bearing of the coefficient of friction and oil film resistance. The results indicated that there was a slight decrease of coefficient of friction, but after approximately 230 minutes it arrived at the equilibrium value. Zero electrical resistance was noted until 245 minutes had elapsed,(but there was no noted change in the friction as the oil film resistance increased) arriving at equilibrium in 400 minutes. Several qualitative runs were made at higher speeds. An illustration of two traces at speeds of 350 opm and 1400 opm is shown in Fig. 19.

Motion pictures were taken of the apparatus and the oscilloscope in the same frame at various speeds. Both resistance traces and friction traces were shown on the scope. For these particular runs an Ellis Strain Analyzer was used instead of the Brush Strain Analyzer used on all the other experiments. Block wiring diagram is shown in Fig. 6.

Except for room temperatures, no other temperatures were taken. Therefore, no direct method of obtaining the dynamic viscosity was available.

DESCRIPTION OF GRAPHICAL RESULTS

Figs. 9 to 11 graphically describe the results obtained in operation of bearing-journal combination number one at various loads and speeds, where:

coefficient of friction

$$f = \frac{\text{lines of displacement at middle of stroke}}{\text{lines per kilogram x load (lb)}} \times \frac{\text{6.6 lb force}}{\text{Kg}}$$

Since the distance from the centerline of the journal to the centerline of the strain gage element is 1.5 inches, and the journal is 1.0 inches in diameter, the value of force read on the recorder is one third as great as that at the surface of the bearing. Therefore, the frictional force at the bearing surface is 6.6 lb. force at bearing surface per kilogram at the strain gage element.

Figs. 12 to 14 show similar results obtained from journal bushing combination number two. Fig. 14 also shows typical recorder traces for the load of 145.5 p.s.i. Notice the similarity of the general shape of the curves. However, the absolute values are generally greater for combination number one.

Figs. 15 to 18 show the results of varying the load on combination number two at a constant speed of 120 opm, and recording simultaneously values of friction and maximum oil film resistance.

Fig. 19 shows two traces on the oscilloscope with both a friction and resistance trace being shown against shaft position. Shutter speed used was 1/2-second; notice the excellent repetition.

Tabulated results will be found in Appendix II.

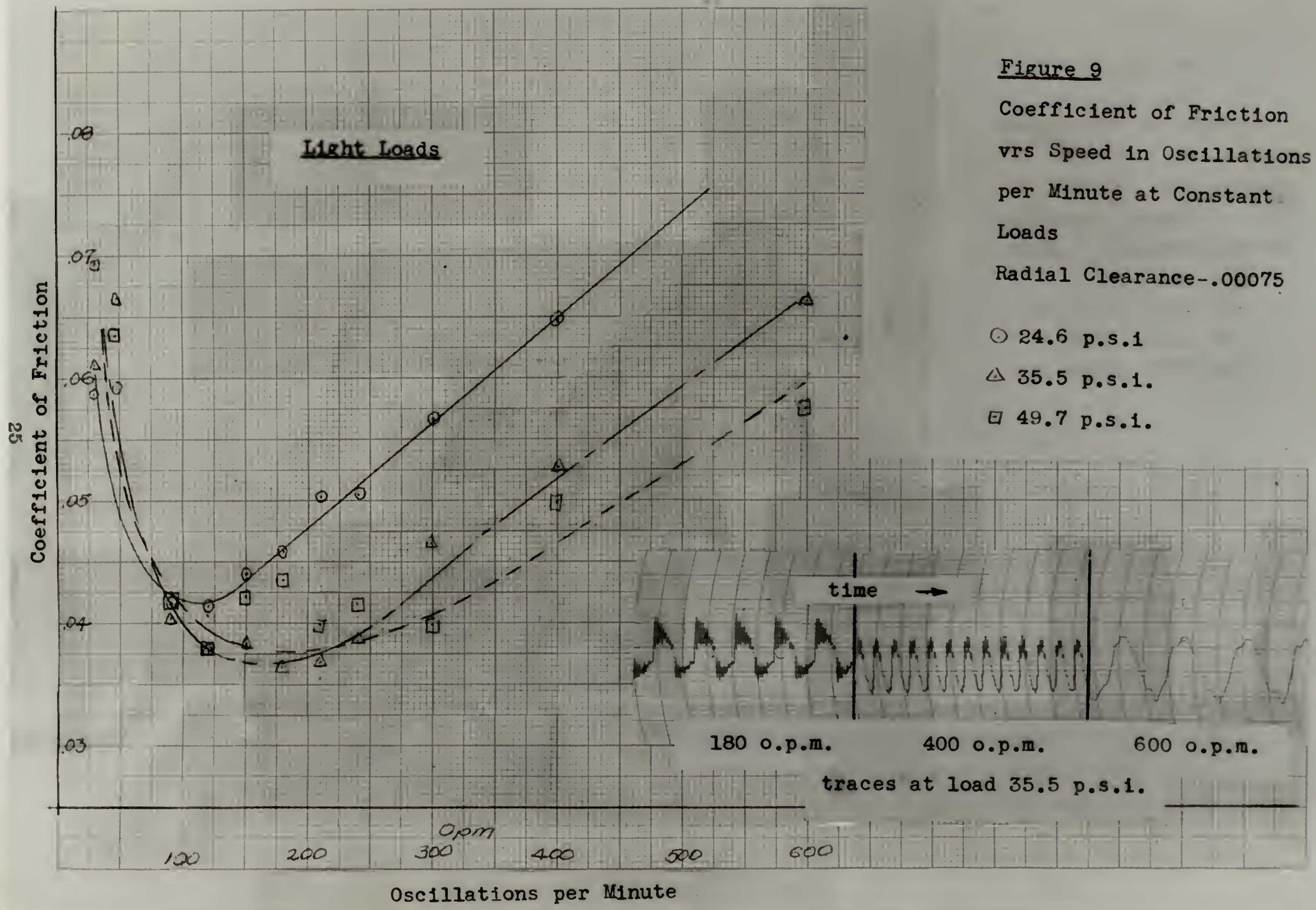


Figure 9

Coefficient of Friction vrs Speed in Oscillations per Minute at Constant Loads

Radial Clearance-.00075

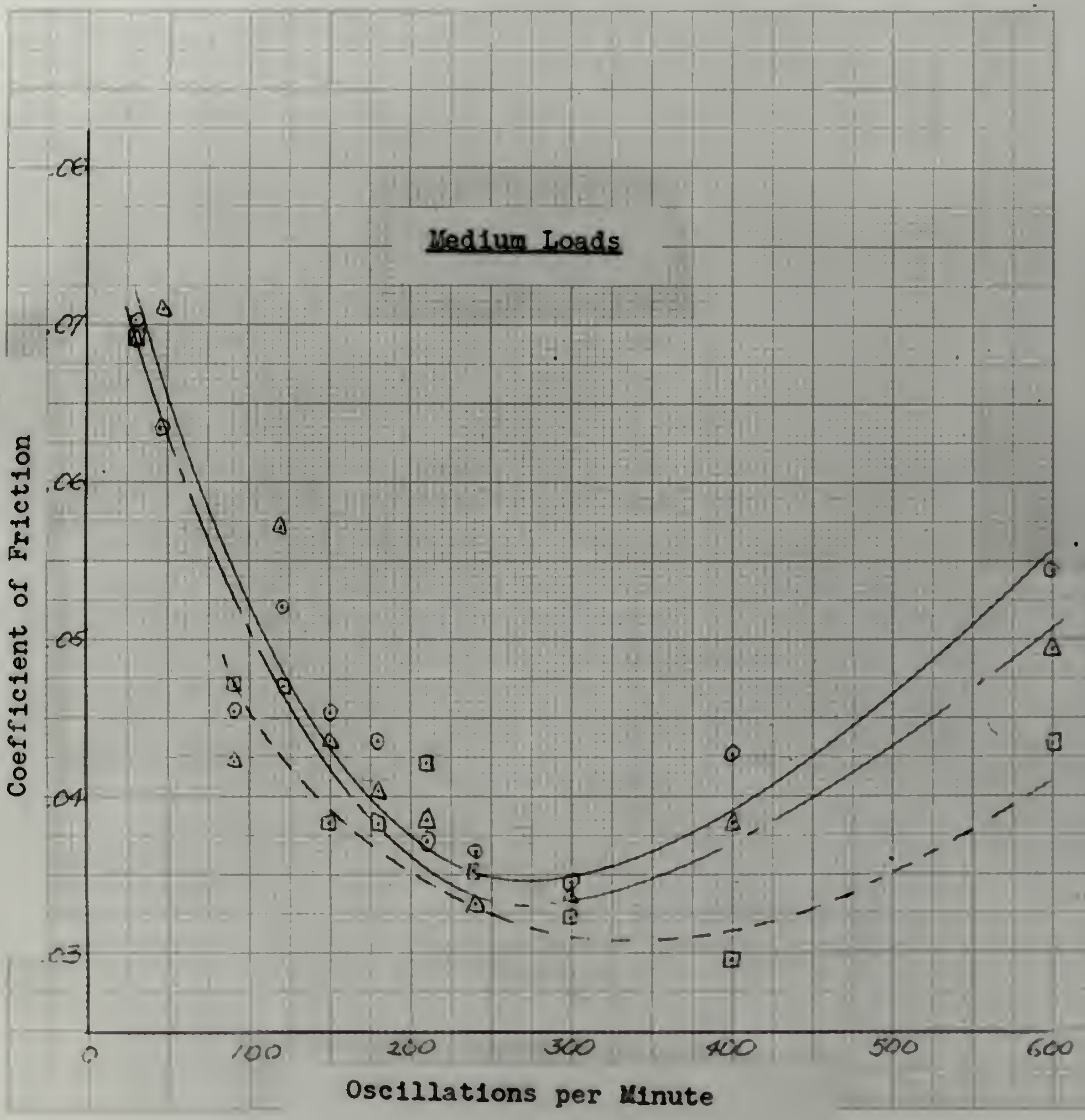


Figure 10

Coefficient of Friction vrs Speed in Oscillations per Minute at Constant Loads.

60.6 p.s.i.

71.5 p.s.i.

93.5 p.s.i.

K+E 10 X 10 TO THE ½ INCH 359-11
KEUFFEL & ESSER CO. MADE IN U.S.A.

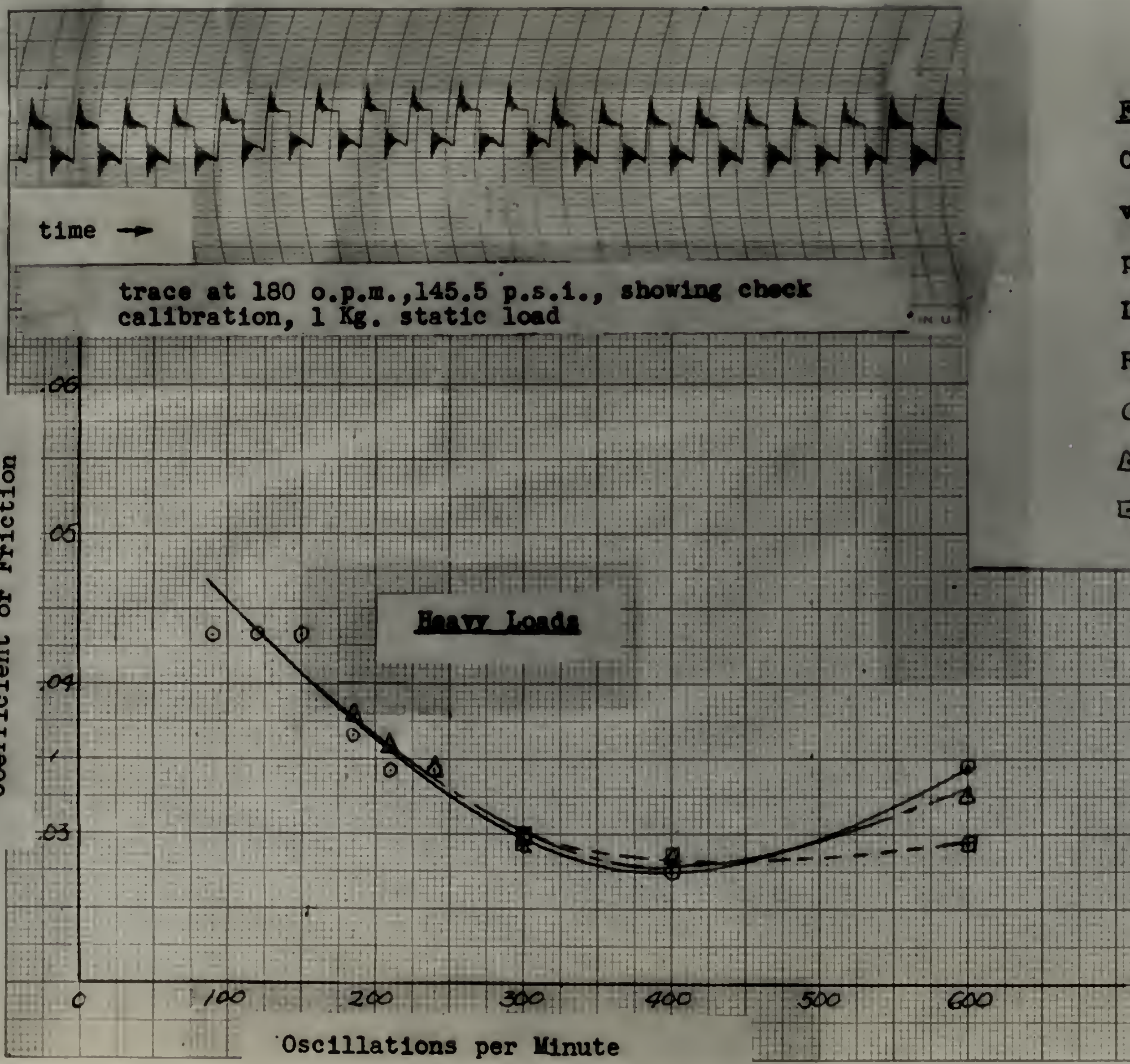

trace at 180 o.p.m.,145.5 p.s.i., showing check calibration, 1 Kg. static load

Figure 11

Coefficient of Friction vrs Speed in Oscillations per Minute at Constant Loads

Radial Clearance-.00075

⊙ 120.6 p.s.i.

△ 145.5 p.s.i.

⊡ 170.5 p.s.i.

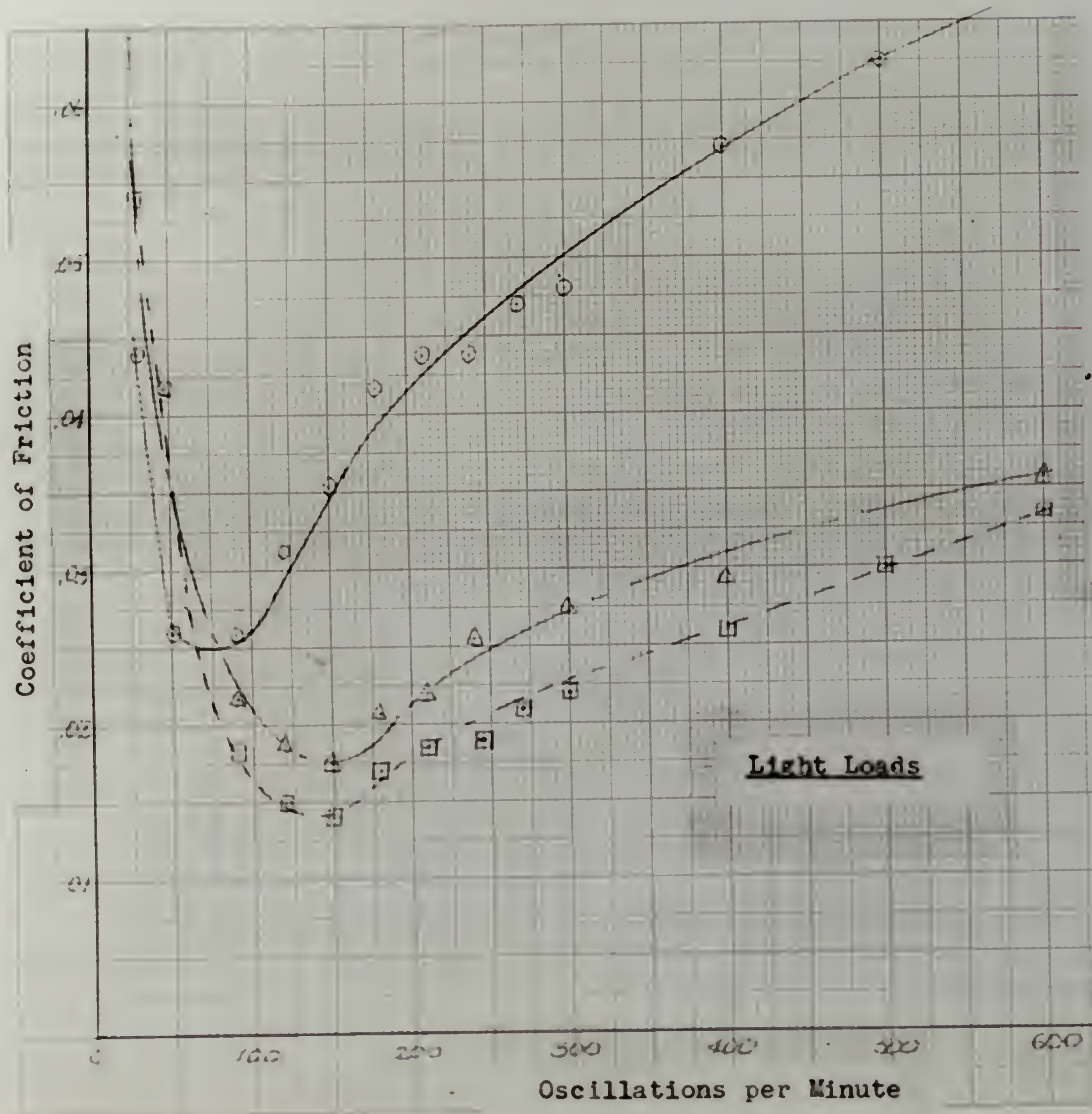


Figure 12

Coefficient of Friction vrs Speed in Oscillations per Minute at Constant Loads.

Radial Clearance-.00045

⊙ 24.6 p.s.i.

△ 49.7 p.s.i.

⊡ 60.6 p.s.i.

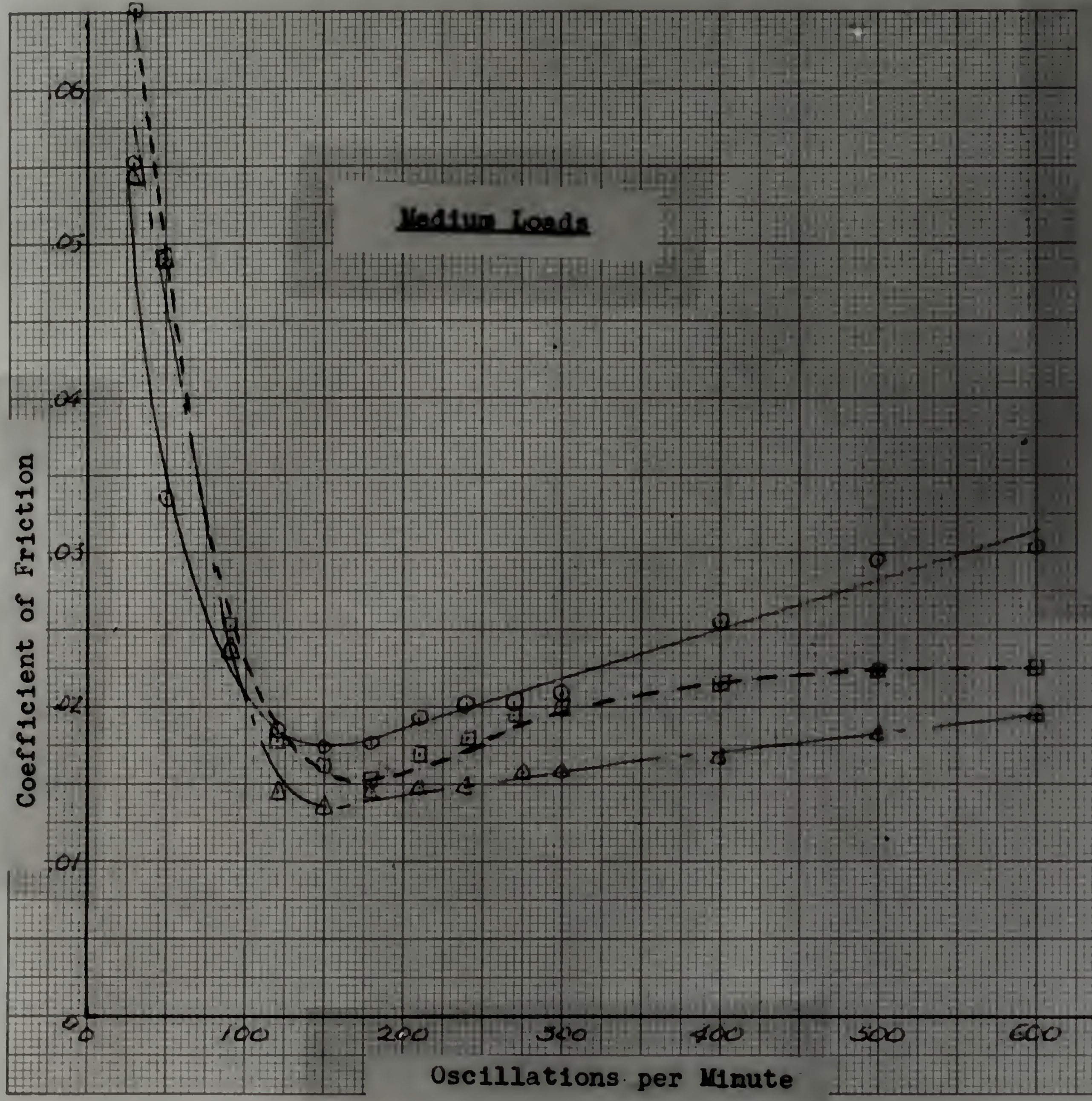


Figure 13

Coefficient of Friction vrs Speed in Oscillations per minute at Constant Loads.

Radial Clearance-.00045

⊙ 71.5 p.s.i.

⊡ 93.5 p.s.i.

△ 120.6 p.s.i.

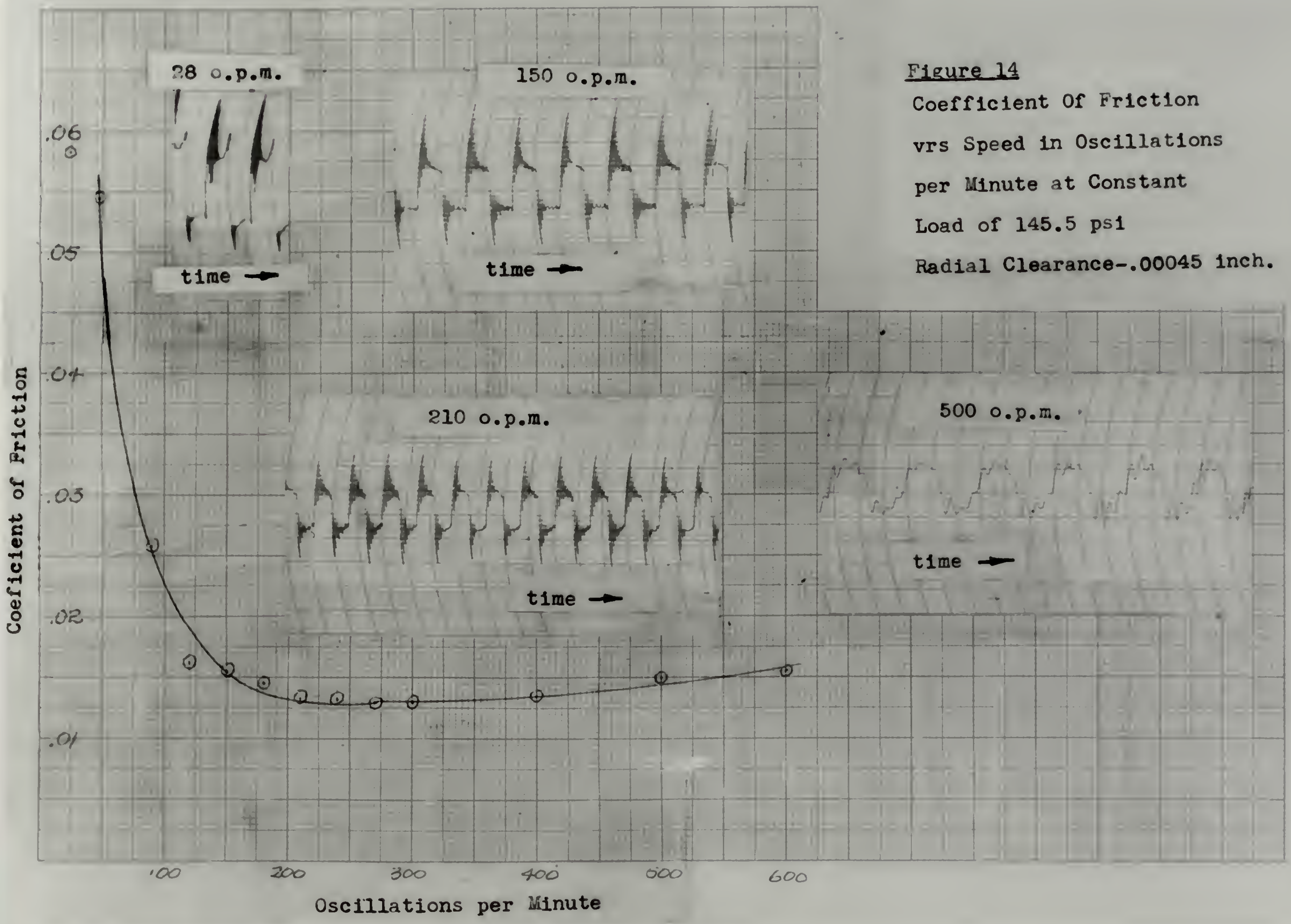


Figure 14
Coefficient Of Friction
vrs Speed in Oscillations
per Minute at Constant
Load of 145.5 psi
Radial Clearance-.00045 inch.

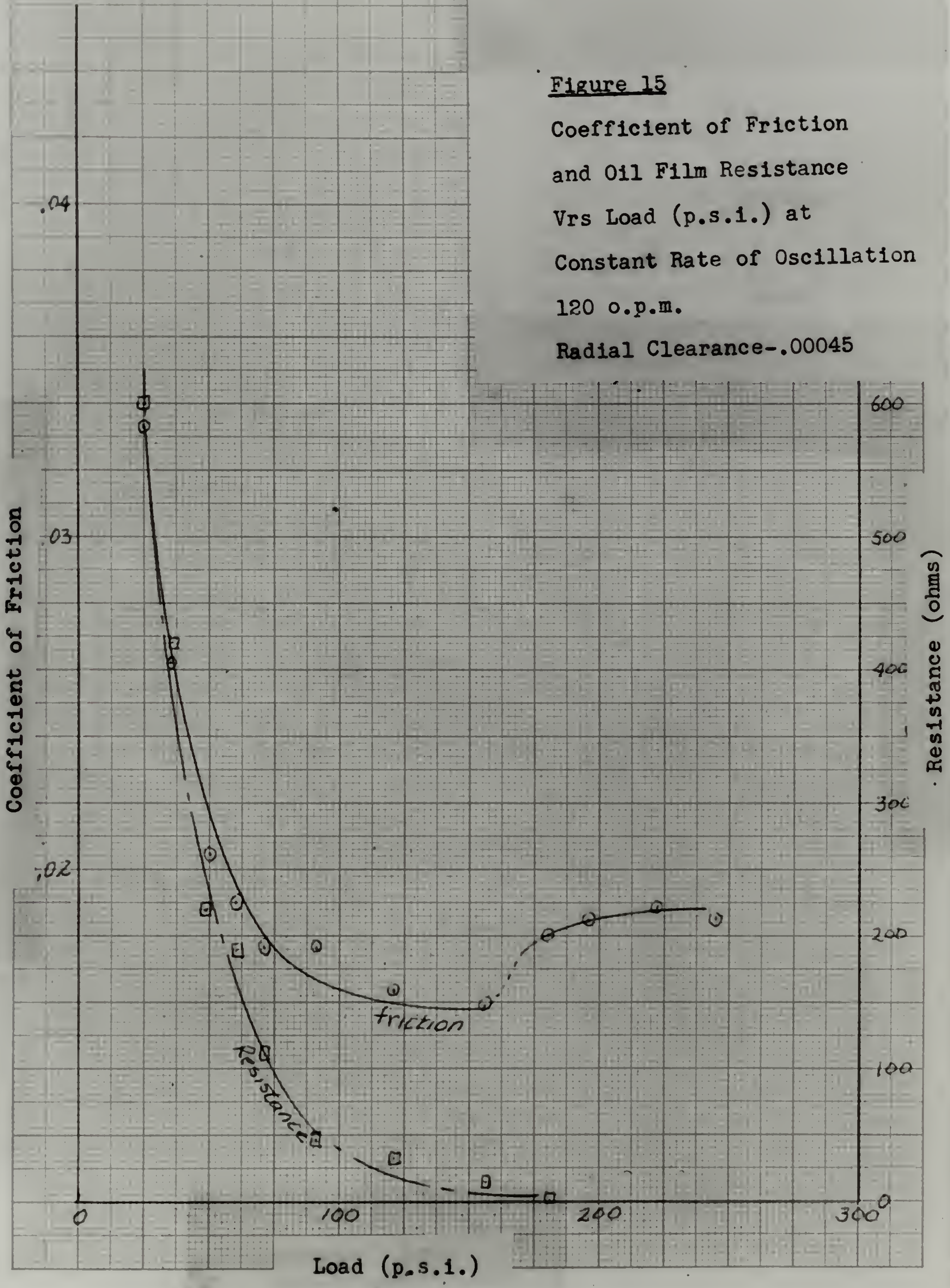
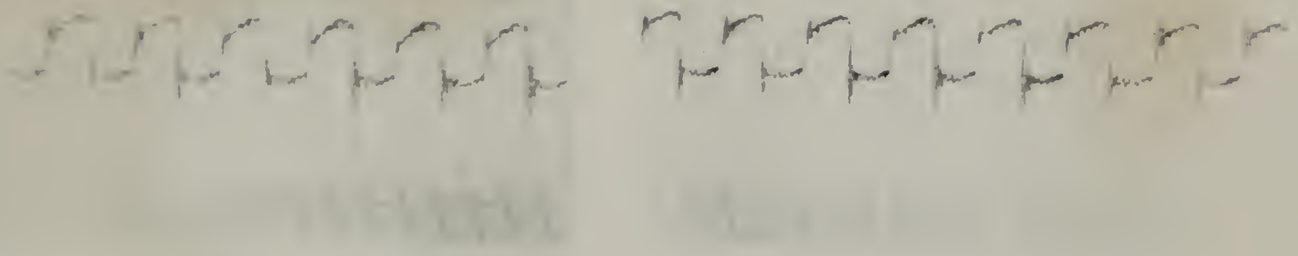


Figure 15

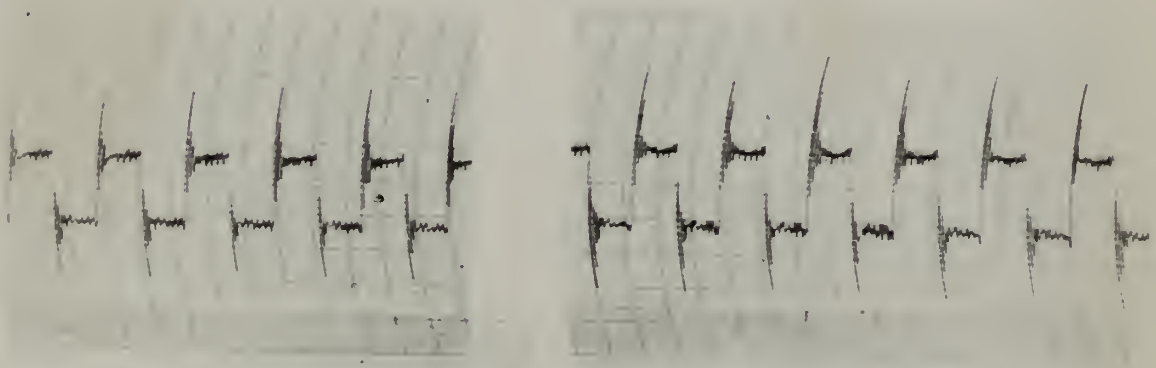
Coefficient of Friction and Oil Film Resistance Vrs Load (p.s.i.) at Constant Rate of Oscillation 120 o.p.m.

Radial Clearance-.00045



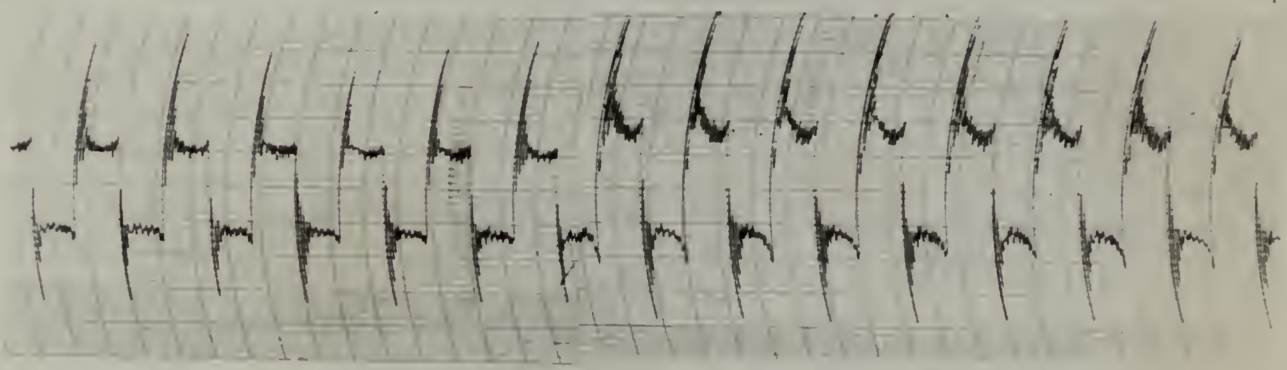
f = .0335 L = 24.6 p.s.i.
(a)

f = .0262 L = 35.5 p.s.i.
(b)



f = .0205 L = 49.7 p.s.i.
(c)

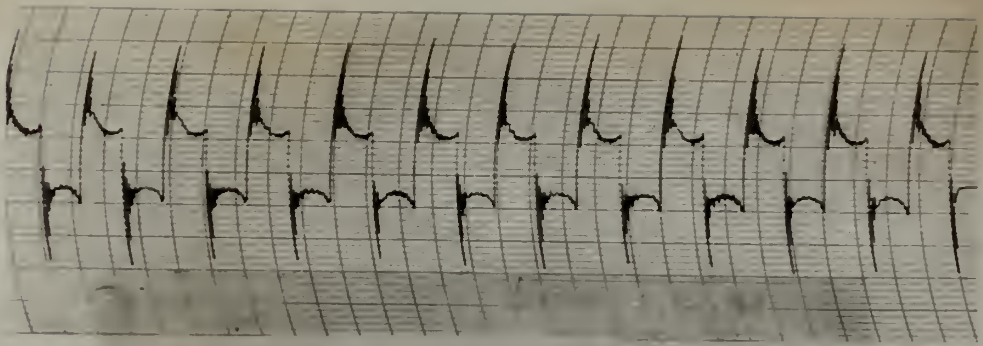
f = .0198 L = 60.6 p.s.i.
(d)



f = .0177 L = 71.5 p.s.i.
(e)

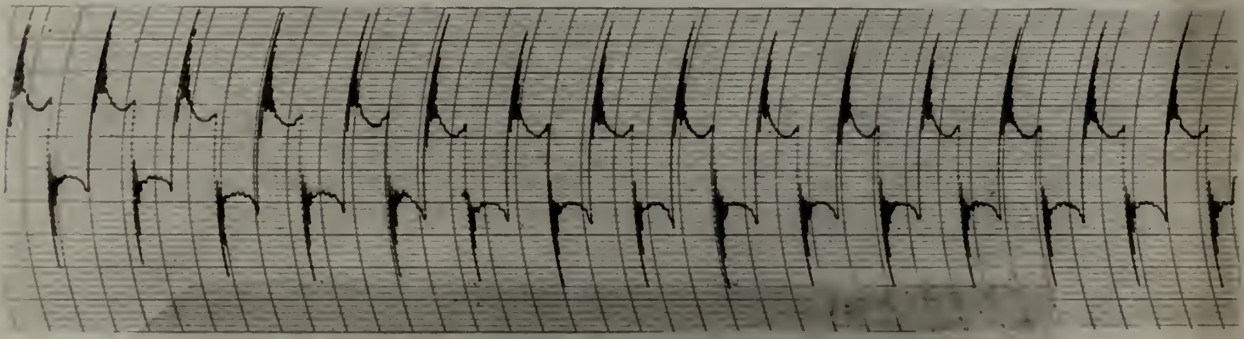
f = .0177 L = 93.5 p.s.i.
(f)

Figure 16 - Friction Traces Versus Time at 120 o.p.m.



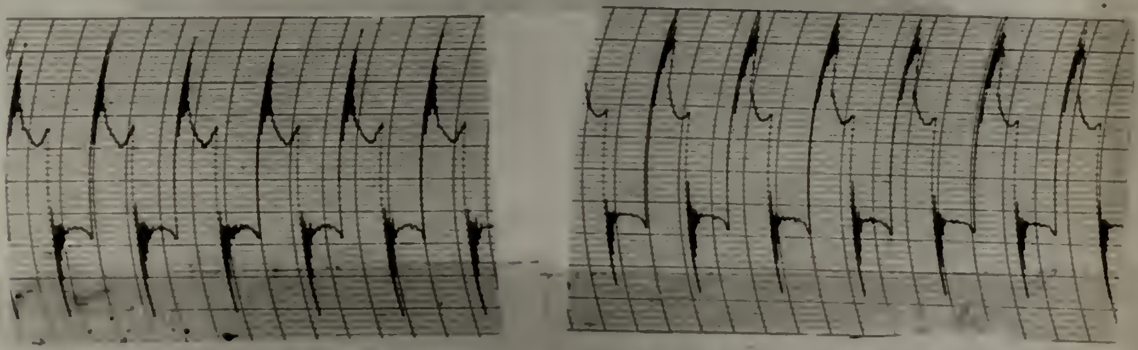
f = .0164 L = 120.6 p.s.i.

(a)



f = .0159 L = 145.5 p.s.i.

(b)



f = .0180 L = 170.5 p.s.i.

(c)

f = .0185 L = 195.5 p.s.i.

(d)

Figure 17 - Friction Traces Versus Time at 120 o.p.m.

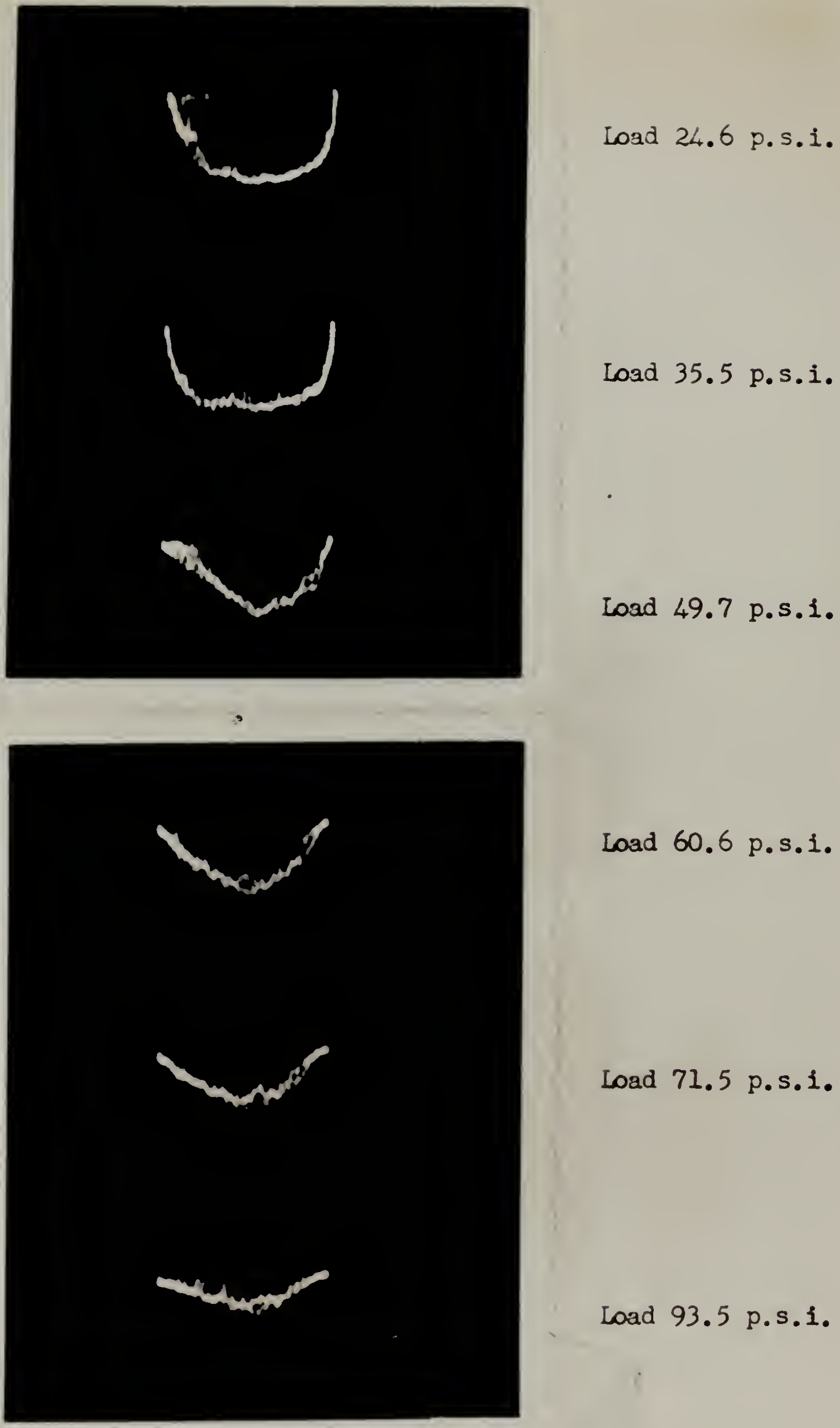
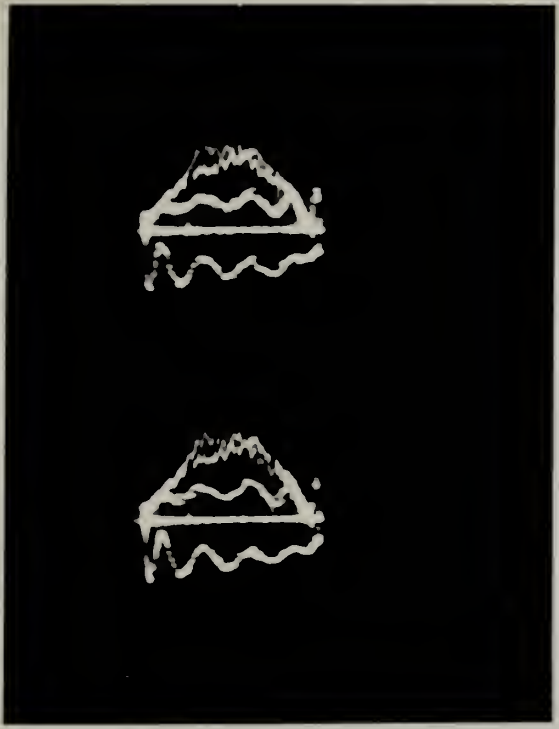
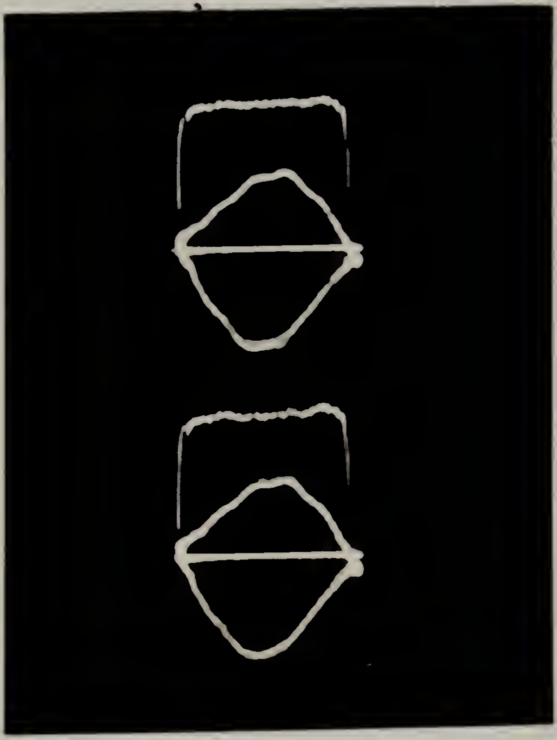


Figure 18. Oscilloscope traces of oil film resistance at 120 c.p.m.



350 o.p.m.
135 p.s.i.



1400 o.p.m.
135 p.s.i.

Figure 19 - Typical Oscilloscope Traces

Straight solid line indicates zero resistance, with increasing value upward. The friction trace is fairly symmetrical about the zero resistance trace. Amplification is the same for both sets of pictures. Note increased value of friction at 1400 o.p.m.

DISCUSSION OF RESULTS

From the results of the series of experiments on this particular bearing undergoing oscillatory motion it is possible to conclude:

(a) fluid lubrication similar to that in a plain rotating sleeve bearing is obtainable with high speeds and/or low load.

(b) four general forms of friction curves versus time were found in oscillating bearing operation, as follows:

(1) sinusoidal
(2) constant value plus sinusoidal
(3) constant value
(4) high points at both ends and lower in the middle.

Two other points which gave indication of possible importance, but which would require the conduct of further research before they could be verified, are:

(a) At constant load with varying rate of oscillation, lower minimum coefficients of friction are obtainable with greater constant loads.

(b) The clearance has a decided effect upon the coefficient of friction.

Since for this investigation, the experiments were conducted with steel journals and bushings, a L/D ratio of one, and a central oil hole, the results can only be duplicated with like geometrical considerations and similar materials. This does not detract from the significance of the results, as the information in general can be applied to other types of materials and grooving.

This is illustrated by the similarity of results between this experiment and Barwell's.

Bowden and Tabor[2], in discussing the use of the electrical resistance measurements across oil films, state that nothing definitely can be said about this phenomena other than that at say 100 ohms an oil film is definitely present. Borge Lunn[5] shows some interesting results, somewhat similar to those obtained in this set of experiments, of oil film resistance measurements made upon a ball rolling across a flat plate with reciprocating motion, but was unable to obtain any quantitative data. It is felt that the results plotted in Fig. 8 definitely point to the fact that with approximately zero oil film resistance, there was a tendency (at 120 opm) for the coefficient of friction curve to flatten out. Also, with increasing oil film resistance, an increase in the coefficient of friction was noted. Because the resistance decreased with load, and simultaneously the coefficient of friction increased, it is felt that his measuring method offered a possible means of determining absolute values of oil film thickness if calibration could be accomplished.

The three general forms of friction curves found by Barwell in his experiments were duplicated as shown in Figs. 9, 14, 16 and 17. The sinusoidal shape illustrated in Figs. 9 and 14 is what Barwell calls a result analogeous to hydrodynamic in sleeve bearing undergoing constant rotary motion. It would be expected that if the oil film were not broken down, and no boundary conditions existed, then $S = Z V/h$. From this it can be seen that

with zero velocity at the end of the stroke, the coefficient of friction, since it is proportional to the shear stress, should be zero, and approach sinusoidal form if $\frac{dh}{dV} \simeq 0$. It should be remarked that the oil film resistance trace gives at most a minimum resistance reading at any point as foreign matter in the lubricant, or metal particles and asperities tend to decrease the resistance reading. The resistance of a perfect journal and bushing should at least have this minimum resistance. It can be conceived that at the ends of the stroke the oil velocity approaches zero, and the particles could conceivably pile up.

A second form which Barwell did not include is illustrated by Fig. 14 at 210 opm, and Fig. 16, (a) and (b). This form, it is believed, could be a combination of hydrodynamic and boundary lubrication. At the beginning of the stroke, a situation can be conceived where a boundary condition exists, that is molecules of oil are trapped or adhere to the surface such that the coefficient of friction is independent of velocity, but as the journal turns the oil film builds up, and the coefficient of friction correspondingly increases as it enters the hydrodynamic regime.

The constant value traces could very well represent a pure boundary condition in which the oil film resistance is not dependent upon velocity. See Fig. 14 at 150 opm, and Fig. 16, (c) and (d).

A type of stick slip phenomena seems to appear in the traces at heavy loads or slow speeds. This seems in general to have the

form of a constant minus a function of shaft position, and could represent a breakdown of the film at the ends of the stroke, and a build up of a type of boundary film in the middle portion of the stroke.

The general curves show some interesting characteristics. First of all the results look very similar to those experimentally obtained from a plain journal bearing under constant rotary motion. The coefficient of friction appears not only to decrease with rate of oscillation for any particular load, but upon reaching a critical speed, to increase. Neither Barwell nor Fogg noted this; Barwell possibly because he never operated at a ZN/P ratio small enough. He did obtain at the lowest load he ran, a flattening out of the curve at constant value of the coefficient of friction. Fogg's experiment did not even approach this regime of operation. Fig. 9 recorder traces illustrate the increasing coefficient of friction with oscillations per minute at 35.5 p.s.i.

In determining the load carrying capacity of a journal bearing, it is normal practice to assume that there are three regimes of operation:

(1) a boundary regime well below the critical point,

(2) a transition zone in the region of criticality where the coefficient of friction is a minimum, and

(3) the hydrodynamic regime, where coefficient of friction increases with speed, or decreases with load.

Normally operation is desired in the region of hydrodynamic lubrication. This seems to be a fair procedure to follow with

this type of bearing, since curves of coefficient of friction versus speed in this region represent not only the friction at the middle of the stroke, but also the maximum value.

BIBLIOGRAPHY

1. F. T. Barwell, A. A. Milne, and J. E. Webber, Some Experiments on Oscillating Bearings, Paper No. 1192, Transactions of Engineers and Shipbuilders in Scotland, 98, pp. 267-326, 1955.

2. F. P. Bowden and D. Tabor, The Friction and Lubrication of Solids, Oxford University Press, 1950.

3. A. Fogg and C. Jakeman, Friction in Oscillating Bearings, Lubrication Research Paper, No. 3, Department of Scientific and Industrial Research, pp. 1-26, 1928.

4. M. D. Hersey and R. B. Snapp, Testing Dynamically Loaded Bearings - I, A Short History of Bearing Test Machines, Trans. ASME, 79, pp. 1247-1259, August 1957.

5. Borge Lunn, Some Experiments with Sleeve Bearing Metals, Transactions of the Danish Academy of Technical Sciences, 2, 1952.

6. W. C. Pierce and E. L. Haenisch, Qualitative Analysis, John Wiley and Sons, 1948.

7. Almo D. Squitero, Iron Plating, Metal Finishing Guidebook Directory, pp. 323-328, Finishing Publications, Inc., 1957.

APPENDIX I

LUBRICATING OIL SPECIFICATIONS

Military Specification Mil-L-15016A 15 August 1950

Type 3050 Lubricating Oil, General Purpose

Viscosity centistokes	(5.75-8.77) at 130°F
SSU at 130°F	45-55
Viscosity Index	75
Pour point	0 (maximum)
Flash point	390 (minimum)
Work factor	.85
Neutrality	Neutral
Neutralization number	.10 (maximum)
Water percent	0
Ash percent	.003
Saponification number	.5 (maximum)
Carbon residue, percent (loose and flaky)	.30 (maximum)
Total sulfur (percent)	.50 (maximum)
Precipitation number	.01 (maximum)

Retained hydrocarbons free from additives, except pour point depressant and anti-foam agents.

APPENDIX II

TABULATED DATA

TABLE I

JOURNAL-BUSHING COMBINATION NUMBER ONE

Radial Clearance .00075 Inches

Load - 24.6 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
10.0	30.0	8.75	1.92	.0782
28.5	16.0	3.50	1.45	.0587
47.0	16.0	3.50	1.45	.0587
90.0	32.0	5.00	1.03	.0419
120.0	16.0	2.50	1.03	.0419
150.0	16.0	2.62	1.08	.0440
180.0	16.0	2.75	1.13	.0461
210.0	16.0	3.00	1.24	.0503
240.0	32.0	6.00	1.24	.0503
300.0	32.0	6.75	1.391	.0567
400.0	15.0	3.75	1.650	.0646
600.0	15.0	5.25	2.310	.0940

Appendix II

TABLE I

(Continued)

Load - 35.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
10.0	15.0	6.50	2.86	.0805
28.5	16.0	5.75	2.37	.0666
47.0	16.0	5.50	2.26	.0637
90.0	16.0	3.50	1.44	.0007
120.0	16.0	3.25	1.34	.0377
150.0	16.0	3.50	1.44	.0407
180.0	16.0	3.50	1.44	.0407
210.0	16.0	3.50	1.44	.0407
240.0	32.0	7.50	1.54	.0436
300.0	32.0	8.00	1.65	.0464
400.0	15.0	4.25	1.87	.0527
600.0	15.0	5.75	2.535	.0714

Appendix II

TABLE I

(Continued)

Load - 49.7 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
10.0	15.0	8.50	3.74	.0752
28.5	15.0	7.25	3.29	.0666
47.0	15.0	6.75	2.96	.0597
90.0	16.0	5.25	2.46	.0435
120.0	16.0	5.12	2.11	.0424
150.0	16.0	5.00	2.06	.0414
180.0	16.0	5.00	2.06	.0414
210.0	16.0	5.00	2.06	.0414
240.0	16.0	5.00	2.06	.0414
300.0	16.0	4.75	1.98	.0398
400.0	14.0	5.25	2.475	.0498
600.0	15.0	6.50	4.33	.0575

TABLE I

(Continued)

Load - 60.6 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
28.5	6.00	3.88	4.26	.0703
47.0	6.00	3.50	3.85	.0635
90.0	6.00	2.50	3.02	.0498
120.0	6.25	3.00	3.165	.0522
150.0	6.00	2.50	2.75	.0454
180.0	6.25	2.50	2.64	.0435
210.0	6.25	2.12	2.24	.0370
240.0	16.0	5.38	2.21	.0364
300.0	15.0	4.75	2.08	.0345
400.0	14.0	5.50	2.59	.0427
600.0	14.0	7.00	3.30	.0544

Appendix II

TABLE I

(Continued)

Load - 71.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
28.5	6.00	4.50	4.95	.0692
47.0	6.00	4.625	5.09	.0711
90.0	6.00	3.75	4.12	.0576
120.0	6.25	3.88	4.09	.0572
150.0	6.00	2.75	3.11	.0435
180.0	6.00	2.62	2.88	.0403
210.0	6.00	2.50	2.755	.0385
240.0	15.00	5.38	2.370	.0331
300.0	15.00	5.50	2.42	.0338
400.0	6.00	2.50	2.75	.0384
600.0	14.00	7.50	3.54	.0494

TABLE I

(Continued)

Load - 93.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
90	6.00	4.0	4.41	.0472
120	6.00	4.0	4.41	.0472
150	6.00	3.25	3.58	.0383
180	6.00	3.25	3.58	.0383
210	6.00	3.12	3.44	.0369
240	6.00	3.00	3.30	.0353
300	6.00	2.75	3.025	.0324
400	6.00	2.50	2.75	.0294
600	13.00	8.00	4.06	.0434

Appendix II

TABLE I

(Continued)

Load - 120.6 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
90	6.00	4.75	5.23	.0433
120	6.00	4.75	5.23	.0433
150	6.00	4.75	5.23	.0433
180	6.00	4.00	4.40	.0367
210	6.00	3.75	4.12	.0342
240	6.00	3.75	4.12	.0342
300	6.00	3.25	3.58	.0297
400	6.00	3.00	3.30	.0274
600	5.75	3.62	4.16	.0345

Appendix II

TABLE I

(Continued)

Load - 145.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
180	6.00	5.00	5.49	.0377
210	3.00	2.38	5.22	.0359
240	6.00	4.50	4.95	.0340
300	6.00	3.88	4.26	.0293
400	3.00	1.88	4.12	.0283
600	5.75	4.12	4.74	.0326

Appendix II

TABLE I

(Continued)

Load - 170.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
180	3.00	2.88	6.32	.0370
210	3.00	2.62	6.17	.0361
240	3.00	2.50	5.50	.0323
300	3.00	2.25	4.96	.0291
400	3.00	2.00	4.40	.0258
600	5.75	4.12	4.74	.0278

Load - 195.6 p.s.i.

300	3.00	2.62	5.77	.0295
400	3.00	2.50	5.50	.0282
600	5.75	5.00	5.75	.0294

Appendix II

TABLE II

JOURNAL-BUSHING COMBINATION NUMBER TWO

Radial Clearance .00045 Inches

Load - 24.6 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
28.5	32	5.25	1.08	.0440
50.0	32	3.25	.666	.0258
90.0	32	3.25	.666	.0258
120.0	32	3.75	.772	.0313
150.0	32	4.25	.870	.0354
180.0	32	5.00	1.03	.0419
210.0	32	5.25	1.08	.0437
240.0	32	5.25	1.08	.0437
270.0	32	5.62	1.15	.0468
300.0	32	5.75	1.18	.0481
400.0	32	6.50	1.40	.0569
500.0	32	7.50	1.54	.0627
600.0	32	8.00	1.64	.0667

Appendix II

TABLE II

(Continued)

Load - 49.7 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
30	30	15.50	3.41	.0686
90	30	4.75	1.04	.0220
120	30	4.25	.930	.0187
150	30	4.00	.877	.0176
180	30	4.75	1.04	.0209
210	30	5.00	1.10	.0220
240	30	5.75	1.26	.0253
300	30	6.25	1.37	.0275
400	30	6.62	1.45	.0292
600	30	3.00	1.75	.0352

TABLE II

(Continued)

Load - 60.6 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
30	30.0	18.0	4.39	.0651
50	13.5	5.0	2.44	.0445
90	27.0	4.5	1.09	.0180
120	27.0	4.12	1.00	.0165
150	27.0	4.00	.975	.0160
180	27.0	4.25	1.04	.0171
210	27.0	4.50	1.09	.0180
240	27.0	4.75	1.16	.0191
270	27.0	5.25	1.28	.0211
300	27.0	5.50	1.34	.0220
400	27.0	6.50	1.58	.0261
500	27.0	7.50	1.82	.0301
600	27.0	8.25	2.01	.0331

TABLE II

(Continued)

Load - 71.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
29	15	9.00	3.96	.0552
47	15	5.50	2.42	.0339
90	15	3.88	1.70	.0258
120	30	6.00	1.32	.0185
150	30	5.75	1.27	.0177
180	30	5.75	1.27	.0177
210	30	6.25	1.37	.0192
240	30	6.50	1.44	.0201
270	30	6.50	1.44	.0201
300	30	6.75	1.48	.0208
400	30	8.25	1.82	.0254
500	30	9.25	2.10	.0294
600	30	9.75	2.15	.0301

TABLE II

(Continued)

Load - 93.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
28	14	13.0	6.12	.0653
48	14	9.50	4.48	.0480
90	14	5.00	2.34	.0251
120	14	3.50	1.65	.0176
150	14	3.25	1.52	.0164
180	14	3.00	1.41	.0151
210	28	6.75	1.58	.0170
240	28	7.38	1.74	.0186
270	28	7.75	1.82	.0195
300	28	8.00	1.88	.0201
400	14	4.25	2.00	.0214
500	28	8.75	2.06	.0221
600	28	9.00	2.12	.0227

TABLE II

(Continued)

Load - 120.62 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
30	5.5	6.50	7.80	.0547
50	14.0	12.50	5.89	.0489
90	14.0	6.00	2.82	.0234
120	14.0	3.75	1.77	.0147
150	14.0	3.75	1.77	.0147
180	14.0	3.62	1.71	.0141
210	28.0	7.38	1.73	.0144
240	28.0	7.50	1.77	.0147
270	28.0	8.00	1.89	.0157
300	14.0	4.00	1.89	.0157
400	14.0	4.25	2.00	.0166
500	14.0	4.75	2.24	.0186
600	14.0	5.00	2.35	.0195

TABLE II

(Continued)

Load - 145.5 p.s.i.

o.p.m.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction
30	5.5	7.00	8.36	.0575
50	5.5	6.50	7.78	.0534
90	14.0	8.00	3.77	.0259
120	14.0	5.00	2.36	.0162
150	14.0	4.75	2.24	.0154
180	14.0	4.50	2.12	.0146
210	14.0	4.12	1.95	.0134
240	14.0	4.12	1.95	.0134
270	14.0	4.00	1.89	.0130
300	14.0	4.00	1.89	.0130
400	14.0	4.12	1.95	.0134
500	14.0	4.62	2.18	.0150
600	14.0	4.75	2.24	.0154

Appendix II

TABLE III

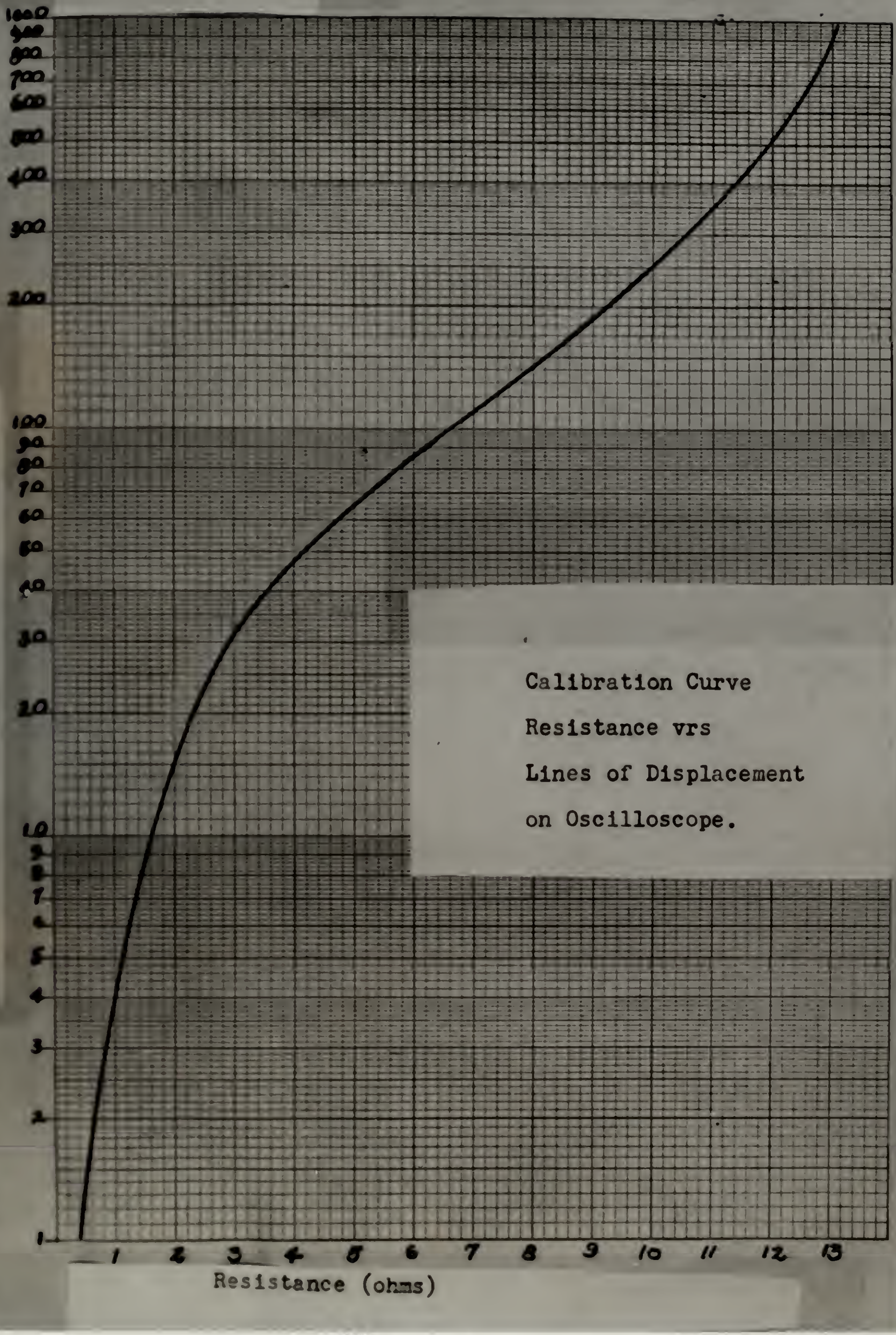
JOURNAL-BUSHING COMBINATION NUMBER TWO

Constant Rate of Oscillation - 120 opm

Radial Clearance - .00045 Inches

Load p.s.i.	Lines/Kg.	Lines	Frictional Force (lb.)	Coefficient of Friction	Oscilloscope Lines	Equivalent Resistance (ohms)
24.6	30	3.75	.824	.0335	12.0	600
35.5	30	4.25	.932	.0262	11.5	420
49.7	30	4.62	1.02	.0205	9.5	220
60.6	30	5.25	1.16	.0198	9.0	190
71.5	30	5.75	1.26	.0177	7.0	110
93.5	30	7.50	1.650	.0177	4.0	47
120.6	15	4.50	1.98	.0164	3.0	33
145.5	15	5.25	2.31	.0159	2.0	15
170.5	15	7.00	3.075	.0180	.8	2.4
195.5	15	8.25	3.63	.0185	---	---
220.6	15	9.50	4.18	.0189	---	---
245.6	15	10.50	4.62	.0185	---	---

Appendix III



Calibration Curve
Resistance vrs
Lines of Displacement
on Oscilloscope.

thesN362
Lubrication study of an oscillating bear
3 2768 002 01808 7
DUDLEY KNOX LIBRARY